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MACHINE DESIGN

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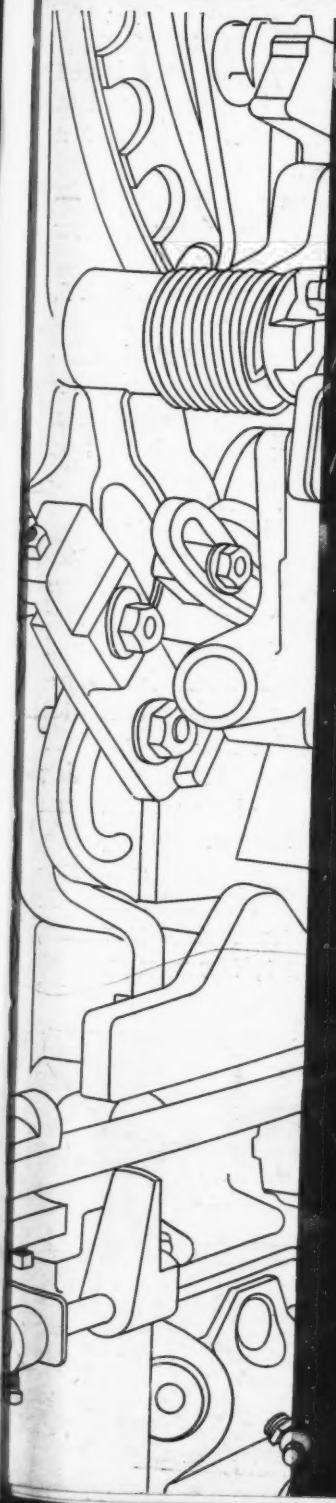

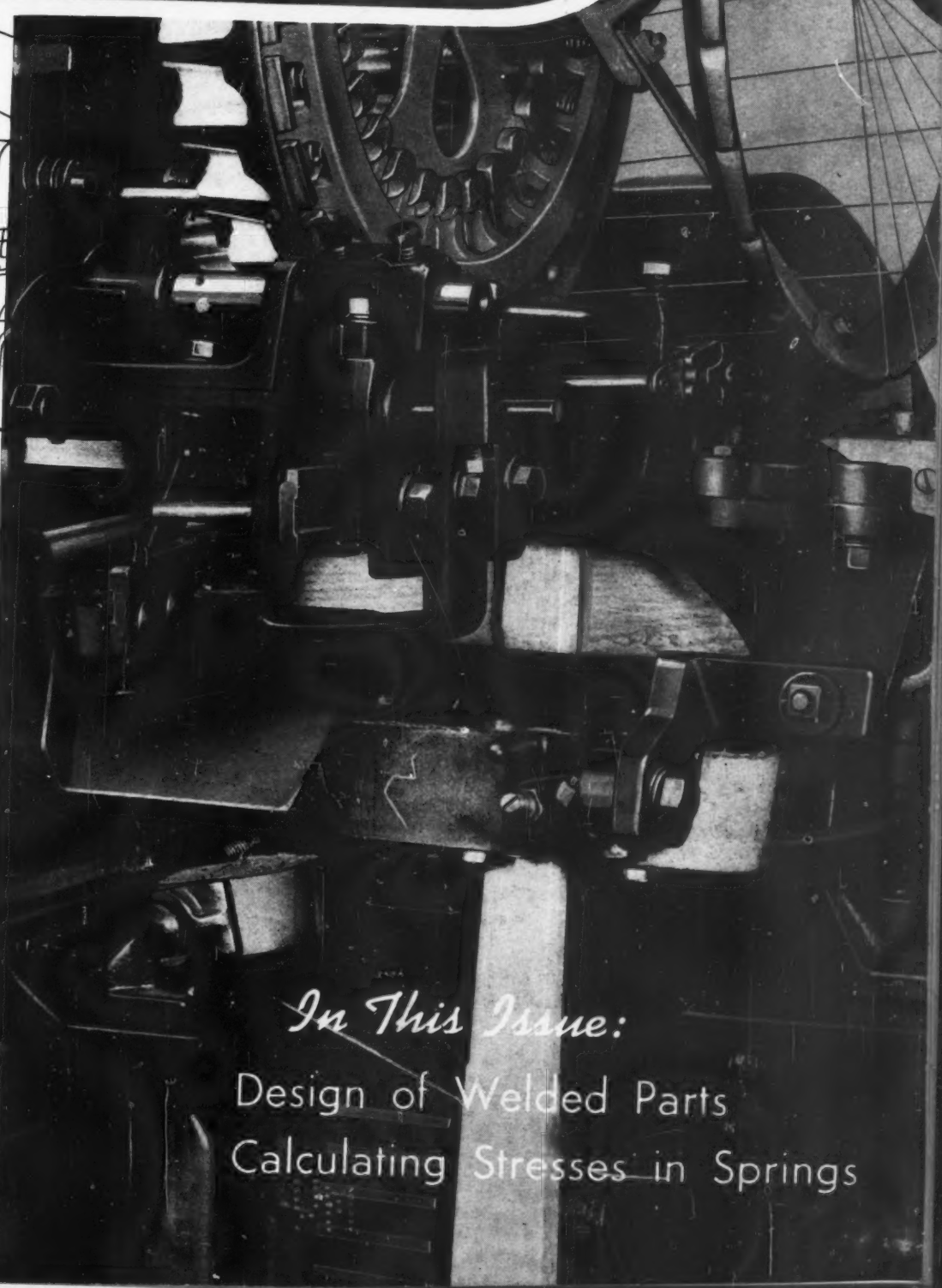

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Design of Welded Parts

Calculating Stresses in Springs

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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

JANUARY, 1945 Volume 17—Number 1

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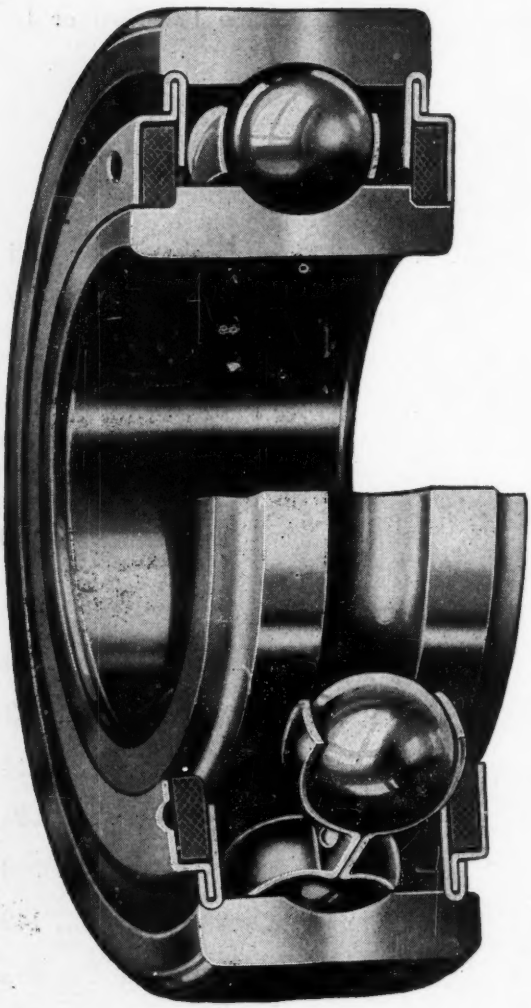

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WRITE FOR BULLETIN 506

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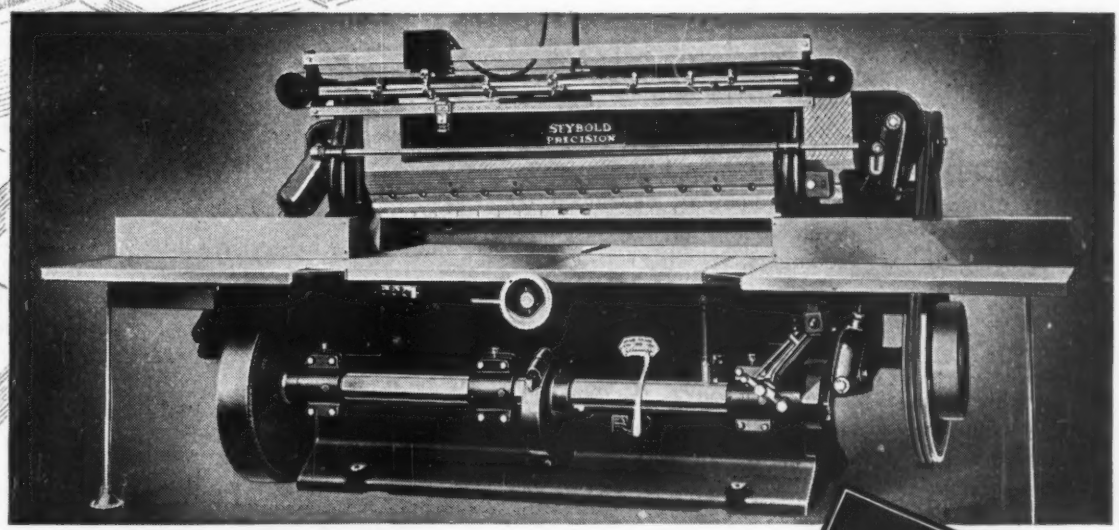
MASSIVE CONSTRUCTION, PRECISION ACCURACY, AND FAST, DEPENDABLE OPERATION characterize the paper and press-board cutting machines made by the Harris-Seybold-Potter Company. Heavy lifts of paper stock and various fibrous materials are often tough to cut. So this equipment is designed to give its knife a cutting force up to 28,200 lbs . . . almost 15 tons.

At every cut a tremendous shock occurs, especially to the power-transmission equipment. Only two Dayton V-Flat Belts handle this load "very satisfactorily . . . giving long, trouble-free service, and requiring surprising minimums of adjustment," says the company.



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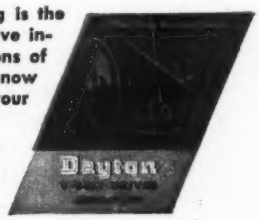

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ADDED THRUST or jet propulsion from the exhaust of turbosuperchargers, developed on the B-23, is being designed into all new airplanes employing turbosuperchargers.

INDICATIVE OF THE POWER developed by the new type jet-propelled planes is the fact that they will climb at speeds faster than other military craft can travel in level flight. In tests their initial rate of climb after a shallow dive is between 25,000 and 30,000 feet per minute.

TRANSOCEAN AIR TRAVEL of the future will be profitable at 3 cents per passenger mile, according to a Brookings Institution study "Civil Aviation and Peace".

TWICE DESTROYED—by the Russians in 1941 and the Germans in 1943—the U.S.S.R. Dnieprostroi hydroelectric plant will be furnished nine 90,000-kva generators. These huge units, already in construction at General Electric, will give the famous generating plant a 15 per cent greater output capacity than it had originally.

NEW GUNSIGHT LAMPS enable gunners to aim directly into the sun with accuracy. Previous gunsights allow aiming within only about 15 degrees of the sun, leaving a dreaded blind spot. The miniature lamp makes the sight lines or reticle visible, obviating the use of dark filters.

TRANSFORMERS NO BIGGER than a fist, yet able to deliver 100 kilowatts for extremely brief intervals, have been designed for special aircraft applications. Production of ultrathin, grain-oriented Hypersil for core laminations has been responsible for this development. The special steel, two mils thick, makes possible the design of high-frequency transformers in which iron cores could not be used previously because of the high losses. On an experimental basis this steel also has been produced in thicknesses of one mil. This is one-seventh the thickness used extensively for 400-cycle aircraft transformers.

AUXILIARY equipment in the Flying Fortress requires 36 kilowatts of generated power as well as ten miles of control wiring.

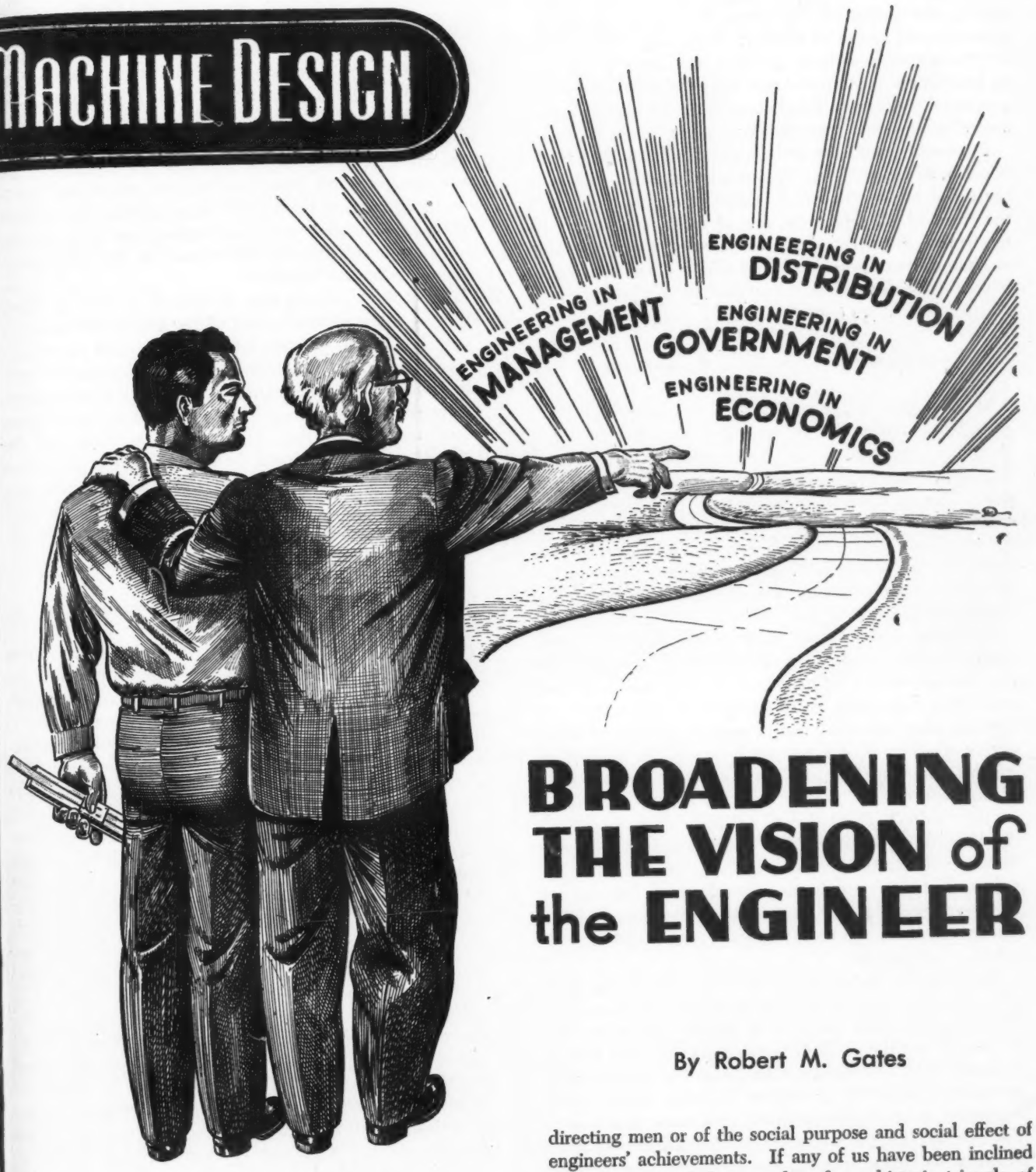
CRITICAL DIGESTS of literature, a useful service performed by the Welding Research Council, provided original sources of information on the welding of armor and special alloy steels and, as a result, welding has been utilized to far greater advantage by us than by our enemies.

TORSION BAR SUSPENSION used on Buick M-18 tank destroyers and Cadillac M-24 light tanks is relatively new for American combat equipment, although such suspensions have been used on Russian and German tanks for a long time. Also they have been used successfully on domestic passenger busses.

RECOGNIZING THE IMPORTANCE of authoritative reference works in the libraries of universities, colleges and technical schools the James F. Lincoln Arc Welding Foundation is establishing in certain institutions a collection of publications on technical phases of welding.

EDISON conducted 6000 experiments, mostly on vegetable growth, for eight years while seeking a suitable incandescent lamp filament. Carbonized bamboo made his lamp possible in 1879. Previously weak lamps had been constructed but a lasting filament had not been devised.

MACHINE DESIGN



BROADENING THE VISION of the ENGINEER

By Robert M. Gates

SOMETIMES it has seemed, in our preoccupation with the control of forces and materials, as if that objective were regarded as the sole concern of the engineer, without consideration of the task of organizing and directing men or of the social purpose and social effect of engineers' achievements. If any of us have been inclined to ignore two of these angles of our historic triangle of engineering responsibility, we must recognize now that the task ahead of us involves the whole triangle.

With this in mind I want to discuss the training and retraining of engineers to meet the challenge of war dislocations, of postwar reconversion, and of the prospect of vastly extended engineering opportunity the world around—in short, the education of the engineers of tomorrow to whom we must pass on the torch. We ex-

From an address delivered at the recent annual meeting of the American Society of Mechanical Engineers, of which Mr. Gates was President for 1944.

pect them to build on the heritage we shall leave to them.

During the past year I have had much correspondence and many talks with people interested in this problem. It has seemed to me that a consensus of engineers, educators, and industrial executives as to engineering requirements of tomorrow includes five general assumptions: 1. Specialization without isolation; 2. Character as well as knowledge; 3. Acquaintance with the whole industrial process; 4. Adjustments in human relations; and 5. Extension of educational opportunity.

Engineers cannot be isolationists, as far as the traditional branches of the profession are concerned. Even in the broad and fundamental branch of engineering which the A. S. M. E. represents, few of us in practice can be mechanical engineers alone; we have to learn something of civil, electrical, chemical, and whatever other branches there may be, in order to do intelligently and competently all the jobs that are entrusted to us.

Engineering Technique in Management

Many engineers are called upon to assume major responsibilities in industrial management. In the years ahead we may expect that the management of industry will be recruited more rather than less from our profession. The application of engineering techniques, of the engineering mind, to a broad field of postwar problems, not only here in America but also in other countries, will surely be needed. With these prospects, isolationism must not accompany the necessary specialization in engineering education.

Engineers cannot carry their technical qualifications around as they would carry other tools in a tool-kit—as if a diploma were a set of tools to be appraised and used with little connection with the owner's personality. A diploma may represent only a certain number of hours during which a person has been exposed to instruction, and his temporary acquisition of enough information out of his textbooks to "pass off" certain courses. Judicious employers don't hire diplomas, or assign responsibility on the basis of diplomas. They value formal training if it has entered into the character of the person who has had it, giving him not only understanding of fundamental facts and principles, but also mental discipline, a habit of straight thinking, and some initiative, resourcefulness, and creative imagination.

Experience of many employers with engineering graduates has convinced them that those who have acquired these characteristics can readily acquire the specialized knowledge for the kinds of engineering jobs that may be assigned them. I would not minimize the importance of specialized problems of modern industry; but I would emphasize the more fundamental importance of a foundation of character. For specialized knowledge, once made available by human research and experience, cannot apply *itself* to these problems; it must be applied with ingenuity, resourcefulness, and imagination, usually also with cooperative adaptability, by *persons* who have those qualities.

Engineers have developed for industry an almost unlimited capacity to produce. Yet industry has found no adequate way to move the mass of actual production, much less potential production, economically and continuously to consumers. The relatively rigid cost of distribution has blocked the channel—limiting effective demand, and so checking the flow of production. The productivity we create demands open channels of distribution. Surely engineering techniques need to be applied there. Our obligation to industry includes the whole process.

We know that the very achievements of our profession in exploring and harnessing the forces of nature have not only brought great benefits to mankind, and opened up endless vistas of further technical progress, but also have made great changes in human relations and thereby created grave economic, social and political problems. We have brought water, steam, and electric power, and countless applications of them, into human service; but we have left largely to others—to businessmen and to politicians—the responsibility for adjustment of the consequent changes in human relations.

Hasn't there been in that adjustment a need of adjusters with more understanding of the forces and materials being brought to bear on human relations, of their possibilities and limitations, of the conditions essential for most efficient use? In a society being so changed by engineering achievement—changed and confused by the changes—isn't there need that the informed, trained, and disciplined minds of engineers be directed to the social ends of their achievements? Or does social responsibility rest only on other groups without that sure footing, analytical skill, and clear perspective which engineering training and experience cultivate—or ought to cultivate?

Extending Educational Opportunity

Men returning from war service who have engineering aptitudes and aspirations—where war experience, in many cases, has involved the use of engineering techniques—need adequate provision for suitable training, open to them according to their individual wants and capacities. Such training will be provided not only in our engineering schools, which are expecting to receive large numbers of returning servicemen, but also in other institutions which can offer technical training and in industries themselves, whose experience in training war workers has been a successful educational adventure. Among these servicemen are many, very many, of the engineers of tomorrow, and our engineering societies and technical institutions should make every possible effort to see that the men who have engineering promise get opportunity for training fitted to their potential service.

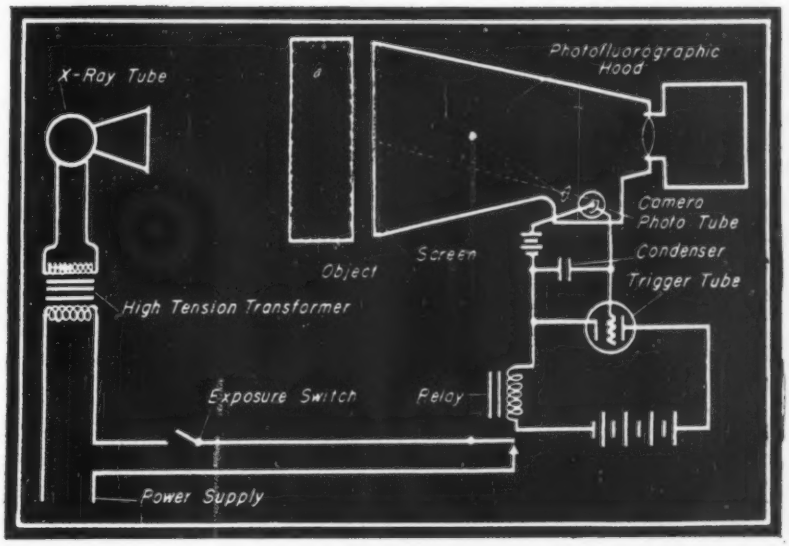
Nor, many of us believe, should we stop there—with the servicemen alone. The movement for assuring wider opportunity for education beyond the secondary-school level has been accelerated during the past few years. The extension of educational opportunity concerns us in engineering, for we want in our profession tomorrow no less than the best our population can furnish.

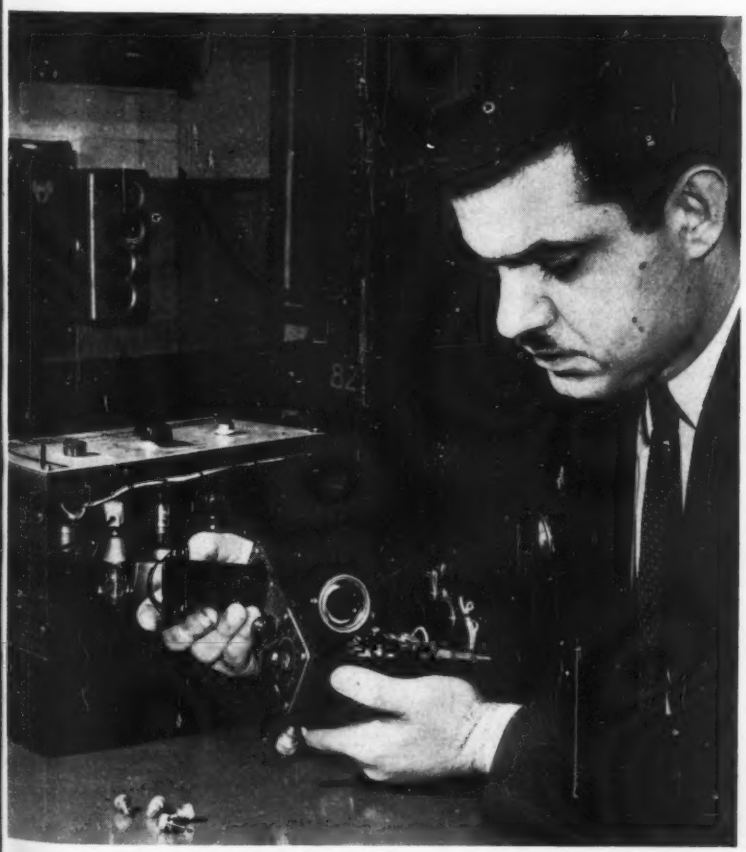
As the horizon widens, as new frontiers open, what engineers will do in the postwar world seems to involve far greater opportunities, far greater responsibilities than ever before. The content and methods and reach of engineering education will inevitably change accordingly. As engineering education keeps pace with expanding opportunities for engineers and the changing requirements of society, so will our profession continue to serve its day and generation with increasing efficiency and with ever-widening fulfillment of its charter of service.

Scanning the Field for Ideas

ELECTRONIC timing will enable radiologists and technicians to obtain uniformly dense X-ray exposures. Shown below, the timer measures the light leaving the fluorescent screen on which the X-rays cast the object image. When sufficient light has left the screen for satisfactory exposure the photoelectric unit actuates a relay to terminate the exposure.

Ingenious simplicity of this timer developed by Westinghouse is revealed by the schematic diagram at right. Closing of the exposure switch energizes the X-ray tube and radiation passes through the object. Undesired,



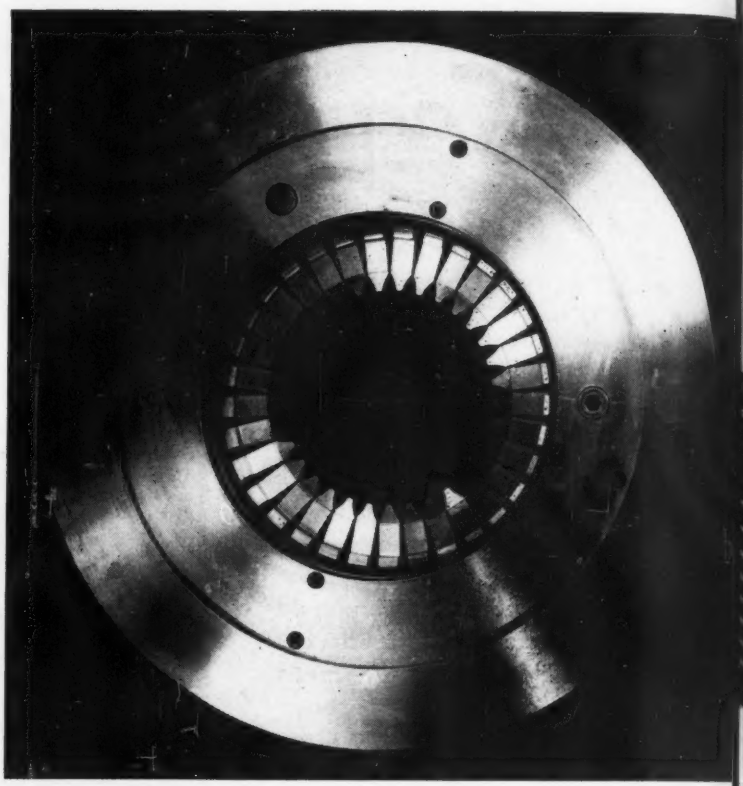


scattered radiations are filtered by a grid and the direct radiation strikes the fluorescent screen where it is converted into visible light radiation. The light picked up by the phototube lens is focused on the tube. Resulting current charges a condenser and when the proper potential is attained a trigger tube fires, actuating the relay and terminating the exposure. The circuit is designed so that the timer maintains a constant exposure factor. Thus, if very dense objects are being photographed the timer maintains the exposure a relatively longer time.

A safety timer—consisting of a trigger tube, an adjustable resistance, a condenser, two relays and a buzzer—protects the X-ray unit against failure of the timer unit and against excessively long exposure times exceeding the capacity of the X-ray tube.

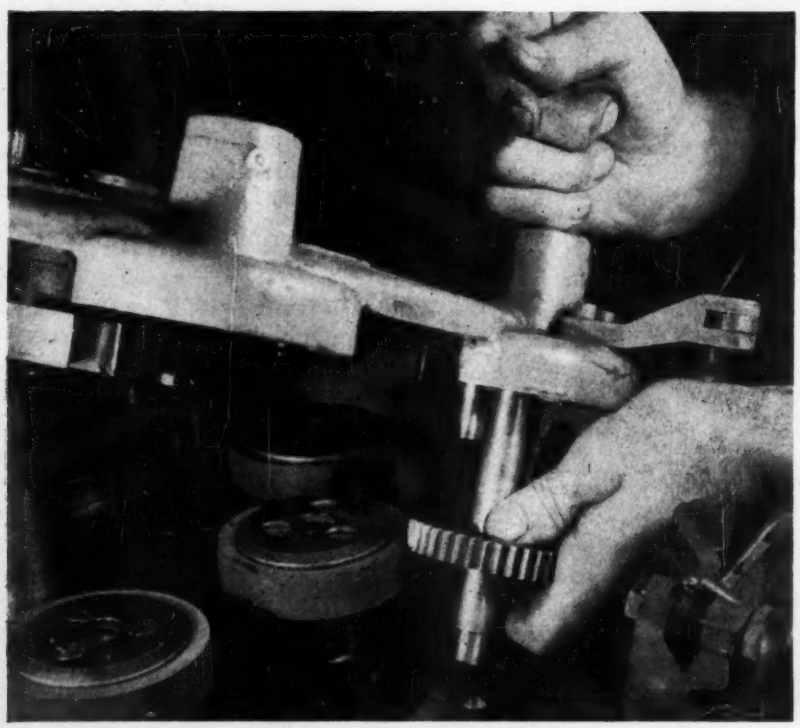
Gear cutting in which all teeth are formed simultaneously by a unique cutter head produces 60 to 100 semifinished

gears per hour. Radially fed, form-tool blades having a shear cutting action progressively cut all gear teeth as the work reciprocates in the machine. At the beginning of each stroke all blades in the cutter head, shown at right, are advanced radially an equal amount. On the return stroke the blades are retracted slightly to provide clearance for the tools. Amount that blades are fed into the work on each stroke is adjustable. Correct sizing of the work is automatically controlled and is also adjustable. The machine on which this type of head is utilized was developed by Michigan Tool Co.



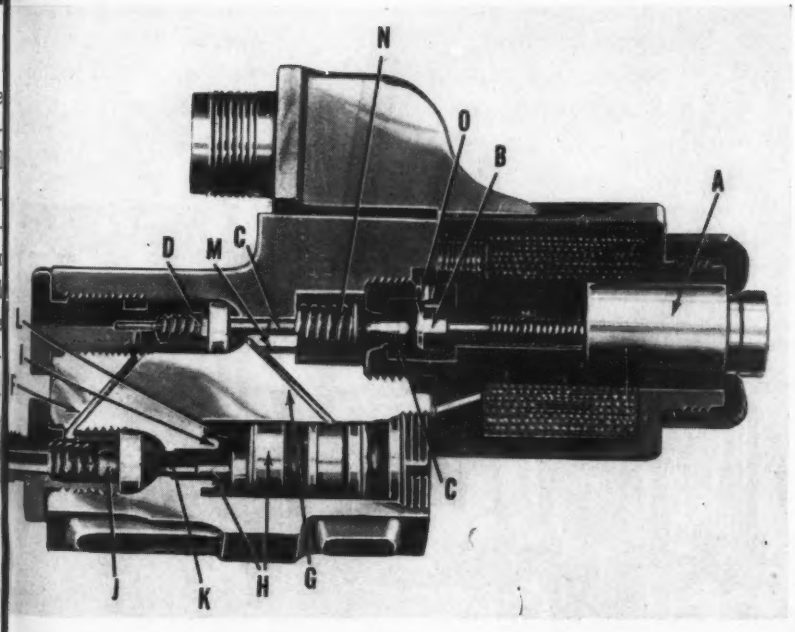
Spindle drive on indexing turret utilizes synthetic rubber driving rings "meshing" with gears on the work spindles as indicated in the photograph below. The turret head is indexed for each of its positions by the geneva mechanism shown at the right in the photograph. As the spindles are advanced to successive stations the geneva provides a dwell while the driving rings rotate the work spindles.

Resilient rings are used to preclude any possibility of the drive jamming when the turret advances. Molded of compar, a vinyl-resin derivative compounded by Resistoflex Corp., the rings have five times the life obtained from rubber. For best results coarse-pitch gears are utilized.



Pneumatic charger clears jammed machine guns in less than one second, obviating laborious manual operations that might be required during critical periods in combat. Developed by Walter Kidde & Co. in cooperation with the Army Air Forces, this two-pound device utilizes compressed air or gas. With a storage cylinder of 57 cubic inches capacity, charged with gas to 1000 pounds per square inch pressure, the charger will operate about 50 times before the gas supply need be renewed. Through use of a special bracket the charger, shown at top of next page, can be demounted or remounted in a few seconds, using only one hand and requiring no tools.

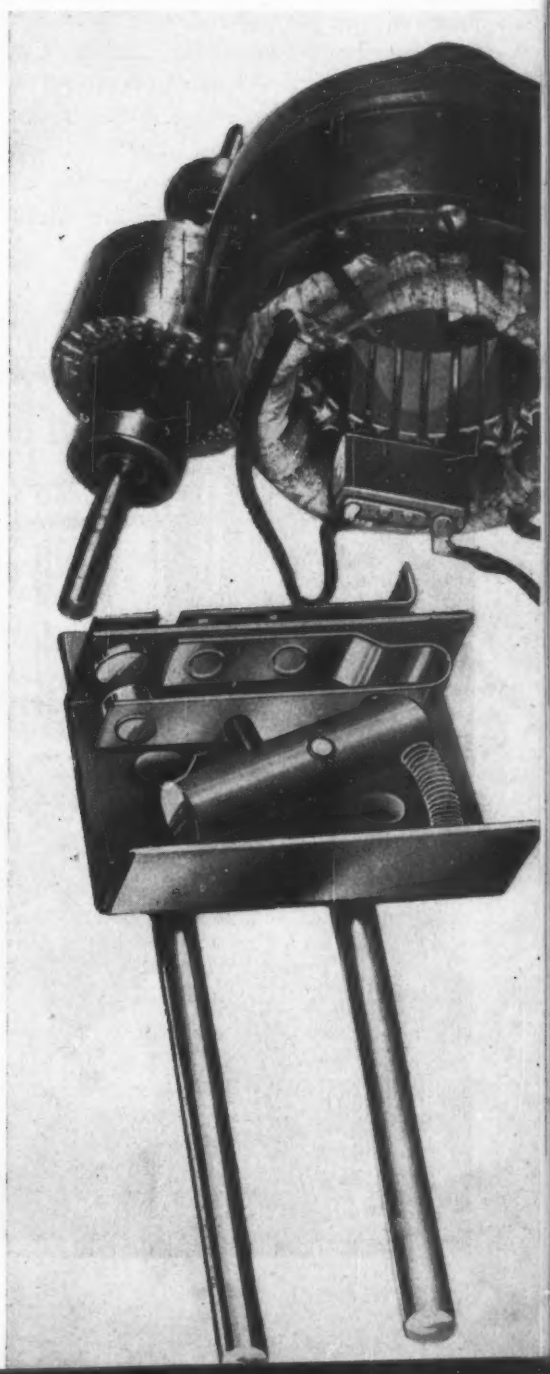
Attached to either side of the gun using the standard gun bolt stud, the charger is actuated by a two-way control valve. This valve may be either a simple manual control or a solenoid valve, remotely controlled. When the solenoid valve, right, is employed, pressure of a button energizes the pick-up coil of solenoid-moving plunger A to the left. A composition cushion B on the plunger stem seals outlet through C while pilot stem moves pilot ball D off its seat, allowing gas to flow from main inlet E through F to passage G. Pressure moves piston H to seat on I, lifting ball J off its seat. This allows gas to flow through outlet K to charger.



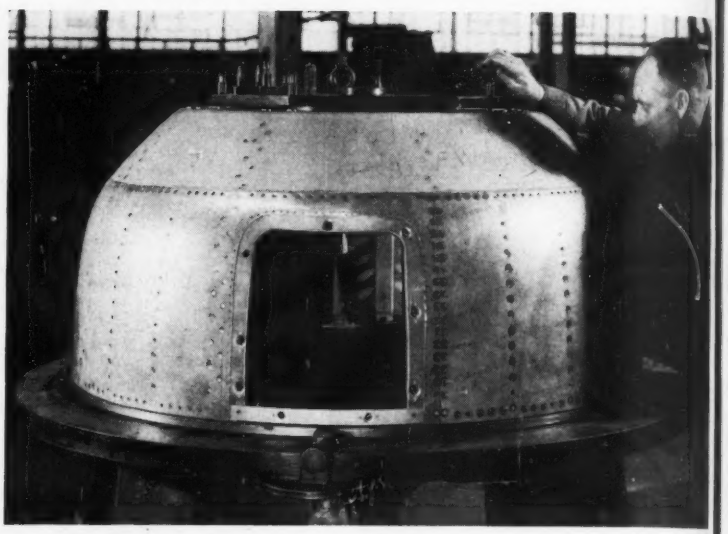
If the holding coil remains energized the system remains statically as described and the charger remains under pressure, holding the gun bolt to the rear out of battery position. If the holding circuit is broken, the valve elements return to position and the charger exhausts through O and L to atmosphere. Two windings are used on the solenoid, one an actuating coil and the other a holding coil, connected in series, drawing 12½ amperes at 28 volts.

Economical use of compressed air is achieved through the system of self-valving. Only the amount of gas necessary to retract the gun action is consumed, automatically shutting off when sufficient air has been used. A slow leak, however, causes the valve to open for compensation. The charger is also utilized to hold the gun in hold-back position to keep shells out of the breech, preventing accidental firing through "cook-off" or shock.

Hermetically-sealed relay, below, is activated by an electromagnet or by magnetic flux such as may be taken from the stator of a split-phase or capacitor-type motor through iron pole pieces inserted between the poles of the motor as shown in the illustration. Action of the flyer in this device, developed by Motor Controls, Inc., loads the contact points by an over-center spring action and absorbs the alternating-current vibration through this loading. Switch may be designed to operate at a given flux density through control of the magnetic pull by increasing or decreasing the air gap between the flyer and its pole pieces and by the spring characteristic opposing



this pull. When applied to a motor, for instance, the inrush of starting current builds up a flux sufficient to close the contacts, energizing the starting winding. When the motor comes up to speed the back electromotive force reduces the flux sufficiently so that the spring-loaded flyer returns to its original position and thus opens the contacts for the starting winding.

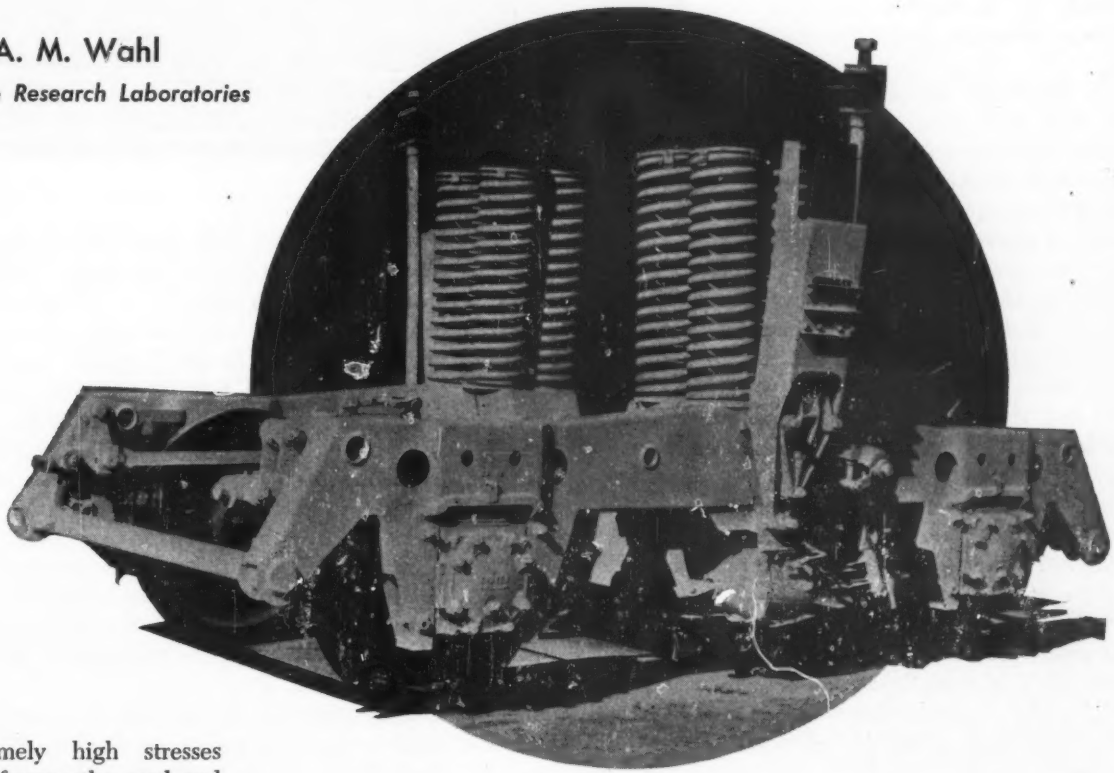


Induction coils in the apparatus shown below are employed to heat internal elements of RCA electron tubes to free metal-absorbed gases from the elements while the tubes are being exhausted. Heat-generating currents are induced in the elements by lowering the high-frequency induction coils over the tubes during the exhausting process. Because no heat is applied from the outside, the glass envelope is unaffected except for heat radiated by the elements from within. Electronic power is utilized to generate the high frequency currents, solving the problem of heating the internal elements without overheating the glass envelope surrounding them.

Changing of engines on attack bombers is facilitated by a new engine mount design, above, which reduces the time required for changing from 18 hours to 4. Designed by Douglas Aircraft, the mount weighs 100 pounds and fits into the nacelle section on the bomber wing. Attached to both the engine and wing at six points, the mount is designed to permit a mechanic to crawl inside through a special door to make repairs and adjustments to the back side of the motor. Braces in regular mounts preclude such servicing.

Utilizing the Effects of Cold Setting in Springs

By A. M. Wahl
Westinghouse Research Laboratories



Cold setting improves efficiency of heavy railway springs as used in pendulum car suspension

THE extremely high stresses which are frequently employed in the design of helical compression springs, particularly those subject to occasional loading only, often seem surprising to many engineers. For example, in music-wire springs of small size, calculated torsion stresses as high as 220,000 to 240,000 pounds per square inch (including curvature correction) may sometimes be used even where the tensile strength is only about 360,000 pounds per square inch for this size material. The use of a high working stress is of great advantage in most spring applications since the energy-storage capacity of the spring, other things being equal, depends on the *square* of the stress. In applications where space is limited, the designer is often forced to use very high working stresses.

***COLD SETTING** has proved advantageous in increasing the life of springs for many types of applications. Analysis of actual springs shows that the reduction of peak stresses may be substantial and indicates why high working stresses may be used in design*

One of the chief reasons why relatively high working stresses can be tolerated in helical compression spring design where fatigue is not involved is that the springs are cold set or scragged (as this operation is sometimes called). This means that the spring is coiled to an initial free length considerably greater than the desired free length. After this is done the spring is loaded with a load far beyond normal working load and beyond the proportional limit. This results in considerable plastic deformation or set, residual stresses being set up which subtract from the load stress. Thus the spring may be stressed higher than would be the case if no cold-setting operation were employed.

It is the primary purpose of the present article to discuss methods for estimating residual stresses set up in helical springs

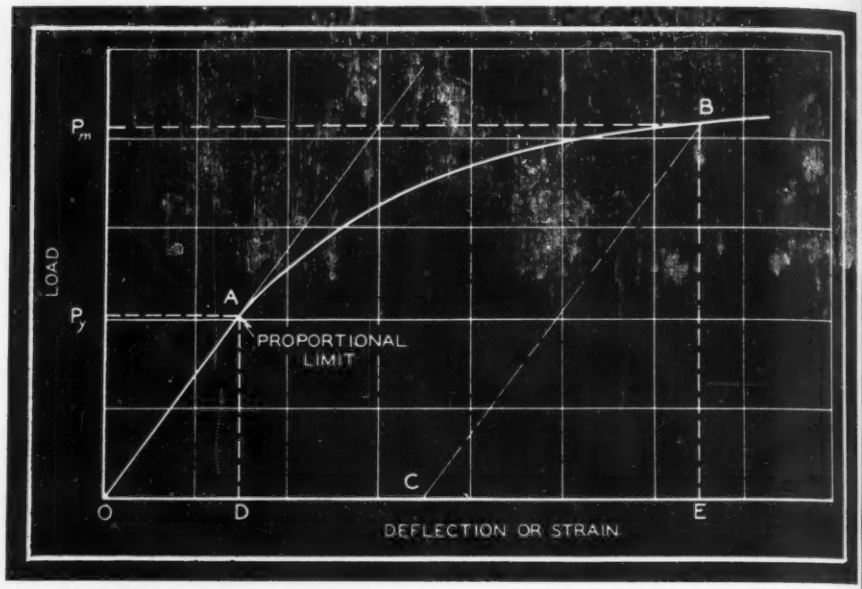


Fig. 1—Typical load-deflection diagram of spring during cold setting

and to determine the reduction in operating stress made possible by the cold-setting operation. Such an analysis will explain why the high working stresses frequently used in practice may be entirely feasible for applications involving static or infrequently applied loading.

It should be noted that while the peak stress in a spring at operating loads may be considerably reduced by cold-setting operations, this will not appreciably affect the *range* of stress in the spring if fatigue loading is present. Since the stress range is the most important criterion with regard to danger of fatigue failure, it may be that the endurance limit of the spring may not be greatly affected by cold setting, although some gain due to reduction in the mean stress of the range may be expected. Also there is evidence that residual or trapped stresses are at least partially lost after a time where fatigue loading is involved. However, where a relatively moderate number of cycles of stress is involved (10,000 to 100,000 cycles) the increase in life due to cold setting may be appreciable. Further test data to check this are needed.

An example of an application in which cold setting improves the performance and efficiency of the springs is the pendulum car suspension shown in the head illustration. In this case, the springs are required to carry not only vertical but also lateral loads in normal operation. By proper cold setting of the springs in this and similar applications, higher working stresses may be used and thus more flexibility may be provided within a given space than would be the case otherwise.

Effects Due to Cold-Setting

A typical load-deflection diagram of a helical spring during the cold-setting or scragging operation is shown in Fig. 1. Between points *O* and *A* the spring behaves practically elastically. At point *A* the elastic or proportional limit in the outer fiber near the inside of the coil is reached; beyond this point yielding and strain hardening of the material occur. If at point *B* the load is reduced, the spring deflection will return approximately along the dot-dash line *BC*. In general, this curve will not be exactly a straight line but it approximates this shape. If the load is now increased to point *B* a small hysteresis loop is usually formed, but after several loadings and unloadings (as is common practice during cold setting) a curve approximating a straight line is reached. It is clear that by means of this process the load at the elastic limit of the spring has been increased from a value equal to P_y at *A* to a value approximating P_m at *B*.

The effect of cold-setting or scragging operations on the stress distribution over the cross section of a spring may be visualized more closely by reference to *Fig.* 2. In this, *Fig.* 2*a* represents the stress distribution over the cross section of a large index spring at a load giving a stress τ somewhat under the yield point, curvature effects being neglected. In the following analysis a large-index spring is assumed but the general conclusions are believed to apply approximately to moderate-index springs also. Assuming for purposes of illustration that no strain hardening occurs, yielding will occur at constant stress τ_y and the distribution of stress during cold setting will be as indicated by the shaded area *oabc* of *Fig.* 2*b*. The nominal stress calculated from the load at cold setting by the usual formula will be equal to τ_n and the stress distribution giving a moment equivalent to that given by the shaded area will be represented by the dashed line *od* in *Fig.* 2*b*.

When the cold-setting load is reduced to zero, a residual or trapped stress distribution as indicated by the shaded area in *Fig.* 2*c* results. This curve is obtained by subtracting the ordinates of the dashed line *od* (*Fig.* 2*b*) from those of the broken line *oab*. The reason for this is that on removing the load the spring will behave approximately elastically; therefore, removal of the load is equivalent to subtracting the elastic stress distribution *od* from the plastic distribution *oab* (*Fig.* 2*b*). This gives the residual stress distribution shown in *Fig.* 2*c*. To find the resultant stress distribution at any load, the nominal load stress distribution is added to that of *Fig.* 2*c*. Thus if a load giving a nominal stress τ corresponding to *Fig.* 2*a* is put on the spring, this equivalent to adding the ordinates of curve *od* (*Fig.* 2*a*) to those of curve *oab* of *Fig.* 2*c* gives the resultant stress distribution *oabc* of *Fig.* 2*d*. It is seen that, due to the cold-setting operation, in this case the peak stress has been reduced from a value τ (*Fig.* 2*a*) to a lower value τ_4 (*Fig.* 2*d*).

In actual springs, this picture will be considerably modified since for most spring materials strain-hardening effects are present, i.e., the line *ab* in *Fig.* 2*b* will tend to rise beyond point *a*. Also the hardness of the spring material particularly for larger bar sizes may be somewhat lower near the center of the bar depending on the hardenability of the material. This would also tend to cause a

rise in the portion *ab* of the curve.

For simplicity, an analysis will first be carried out by assuming yielding at constant stress, i.e., no strain hardening. Later the analysis will be extended to take strain-hardening effects into account.

Analysis of Stress for No Strain Hardening

For a large-index spring which may be considered essentially a straight bar under torsion, where yielding occurs at constant shear stress τ_y (no strain hardening), the stress distribution over the cross section at a given cold-setting load is shown in curve *oab* of *Fig.* 3. In this case yielding has progressed inward to a diameter d_1; this means that the strain at the outer fiber will be in the ratio d/d_1 to the strain at which yielding begins. Letting this ratio be equal to γ and referring to *Fig.* 1, this means that at this point during cold setting (neglecting curvature) the deflection is equal to γ times the deflection *OD* at which yielding first begins.

To calculate the load P_m at which the stress distribution of *Fig.* 3 will be obtained, the following equations are used; these being based on the assumption that the spring cross section is essentially a round bar under torsion. The torque $P_m r$ acting on spring cross section is (1)*

$$P_m r = \tau_y \int_0^{\frac{d_1}{2}} \frac{2\rho}{d_1} 2\pi\rho^2 d\rho + \tau_y \int_{\frac{d_1}{2}}^{\frac{d}{2}} 2\pi\rho^2 d\rho \quad \ldots\ldots (1)$$

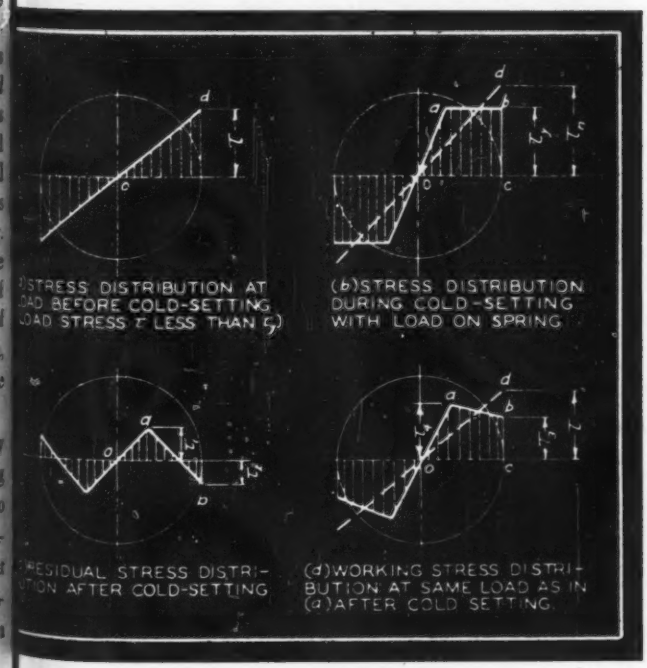
In this, r = mean coil radius, ρ = radius from center O to any point on the spring bar cross section.

Integrating Equation 1 and solving for the load P_m

$$P_m = \frac{\pi d^3 \tau_y}{16r}\left[\frac{4}{3}\left(1-\frac{1}{4\gamma^3}\right)\right] \quad \ldots\ldots (2)$$

*Numbers in parentheses refer to references at end of article.

Fig. 2—Below—Effect of cold setting on stress distribution over cross section of helical spring


(b)STRESS DISTRIBUTION DURING COLD-SETTING WITH LOAD ON SPRING
(d)WORKING STRESS DISTRIBUTION AT SAME LOAD AS IN (a)AFTER COLD SETTING.

In this, as previously mentioned

$$\gamma = \frac{\text{strain at outer fiber}}{\text{strain at which yielding starts}} = \frac{d}{d_1}$$

Also since strain is proportional to deflection, the ratio γ may be taken as

$$\gamma = \frac{\text{deflection at load } P_m}{\text{deflection at which yielding starts}}$$

Thus in *Fig.* 1, at load P_m, γ = ratio *OE/OD*.

If P_y = load at which yielding starts (point *A*, *Fig* 1) then, from the usual formula, neglecting curvature

$$P_y = \frac{\pi d^3 \tau_y}{16r} \quad \ldots\ldots (3)$$

Using this in Equation 2,

$$P_m = \frac{4}{3} P_y \left(1-\frac{1}{4\gamma^3}\right) \quad \ldots\ldots (4)$$

For large deflections, i.e., large values of γ, this indicates that for *no strain hardening* the maximum load will be equal to 4/3 the load at which yielding starts (2).

The nominal or equivalent stress τ_n at load P_m is (from the usual formula)

$$\tau_n = \frac{16 P_m r}{\pi d^3} \quad \ldots\ldots (5)$$

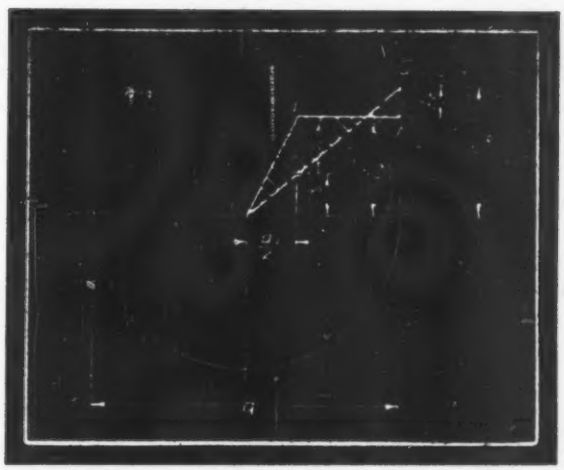


Fig. 3—Stress distribution during cold setting, no strain hardening. Residual stress is represented by shaded area

Solving this for P_m and substituting in Equation 2, the nominal stress τ_n becomes

$$\tau_n = \frac{4}{3}\tau_y \left(1-\frac{1}{4\gamma^3}\right) \quad \ldots\ldots (6)$$

Referring to *Fig.* 3 the shaded area between curve *oab* and line *od* represents residual stress. At point *a* the residual stress τ_1 is equal to $\tau_y - \tau_1'$ where $\tau_1' = \tau_n d_1/d = \tau_n/\gamma$.

This gives for the residual stress τ_1

$$\tau_1 = \tau_y - \frac{\tau_n}{\gamma}$$

Using Equation 6 for τ_n in this, the residual stress at point *a* becomes

$$\tau_1 = \tau_y\left[1 - \frac{4}{3\gamma}\left(1 - \frac{1}{4\gamma^3}\right)\right] \quad \ldots\ldots (7)$$

Similarly the residual stress τ_2 at the outer fiber (point *d*, *Fig.* 3) is equal to $\tau_n - \tau_y$. Using Equation 6 in this the stress τ_2 becomes

$$\tau_2 = \frac{\tau_y}{3}\left(1 - \frac{1}{\gamma^3}\right) \quad \ldots\ldots (8)$$

Assuming that a working load *P* is put on the spring, the stresses due to load *P* will be represented by the dashed

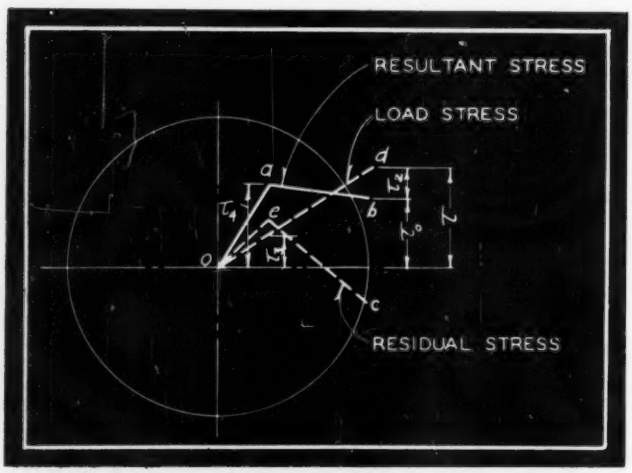


Fig. 4—Resultant stress distribution at load, no strain hardening

line *od* in *Fig.* 4 and must be added to the residual stresses present as a consequence of cold setting, represented by lines *oec*. Letting τ = nominal stress at the outer fiber due to load *P*, then neglecting curvature effects,

$$\tau = \frac{16Pr}{\pi d^3} \quad \ldots\ldots (9)$$

The stress τ_3 at point *a* (*Fig.* 4) due to the load *P* will then be $\tau d_1/d$ or τ/γ. The resultant stress at point *a* becomes

$$\tau_4 = \tau_3 + \tau_1$$

Using Equation 7 in this, the resultant stress at *a* becomes

$$\tau_4 = \frac{\tau}{\gamma} + \tau_y\left[1 - \frac{4}{3\gamma}\left(1 - \frac{1}{4\gamma^3}\right)\right] \quad \ldots\ldots (10)$$

At the outer fiber (point *b* in *Fig.* 4) the stress will be $\tau_0 = \tau - \tau_2$. Using Equation 8 in this,

$$\tau_0 = \tau - \frac{\tau_y}{3}\left(1 - \frac{1}{\gamma^3}\right) \quad \ldots\ldots (11)$$

The resultant stress distribution is thus represented by line *oab*. It should be noted that since strain hardening was neglected, the value of τ_4 given by Equation 10 is probably high while that of τ_0 is probably low. A method of taking strain-hardening effects into account will be discussed in the next section.

Analysis Including Strain-Hardening Effects

In most actual springs, after the yield point has been passed, because of strain-hardening effects the stress will increase as the strain increases. As a first approximation, for analytical purposes it will be assumed that the stress distribution over the cross section above the yield point may be represented by the broken line *oab* in *Fig.* 5. Actually the portion *ab* of this line will be somewhat curved depending on the material and amount of strain, but the assumption of a straight line should be sufficient for the present purpose.

Letting P_m = load during cold setting giving a stress distribution as shown by line *oab* in *Fig.* 5, the stress τ' at any radius ρ less than $d_1/2$ will be equal to

$$\tau' = \frac{2\tau_y\rho}{d_1} \quad \ldots\ldots (12)$$

The stress τ'' at a radius ρ greater than $d_1/2$ will be equal to

$$\tau'' = \tau_y\left(1 + \frac{2\rho - d_1}{d_1 k}\right) \quad \ldots\ldots (13)$$

where

$$k = \text{ratio}\frac{\text{slope of curve oa}}{\text{slope of curve ab}}$$

The ratio *k* may be considered as the ratio of the slopes of the stress-strain curve before and after reaching the

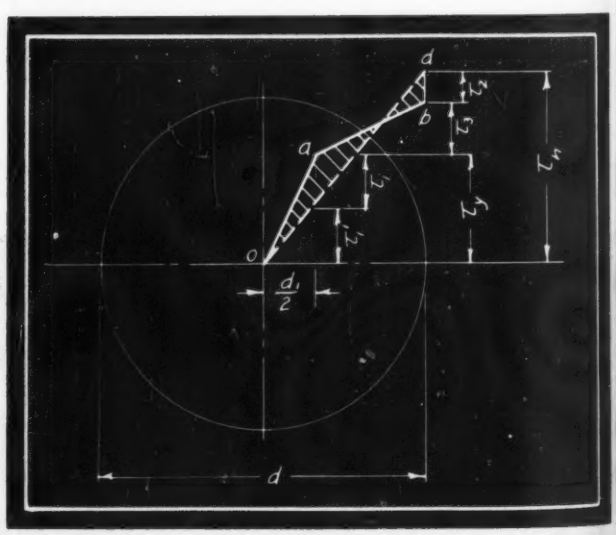


Fig. 5—Stress distribution during cold setting with strain hardening. Residual stress is represented by shaded area

yield point. Low values of *k* correspond to high values of strain hardening and vice versa.

The torque carried by the section at a load P_m will be

$$P_m r = \int_0^{\frac{d_1}{2}} 2\pi\tau'\rho^2 d\rho + \int_{\frac{d_1}{2}}^{\frac{d}{2}} 2\pi\tau''\rho^2 d\rho$$

Using Equations 12 and 13 for τ' and τ'' in this load P_m becomes

$$P_m=\frac{\pi d^3\tau_y}{16r}C_1 \quad \ldots\ldots(14)$$

where

$$C_1=\frac{4}{3}\left(1-\frac{1}{k}\right)\left(1-\frac{1}{\gamma^3}\right)+\frac{1}{\gamma^3}+\frac{\gamma}{k}\left(1-\frac{1}{\gamma^4}\right) \quad \ldots\ldots(15)$$

For $k=\infty$ (no strain hardening) Equation 14 reduces to Equation 2.

The nominal stress τ_n (*Fig.* 5) due to a load P_m is equal to $16P_mr/\pi d^3$ as before. Using this in Equation 14,

$$\tau_n=C_1\tau_y \quad \ldots\ldots(16)$$

As before the residual stress is represented by the shaded area (*Fig.* 5) between the actual stress distribution curve *oab* and the straight line *od* representing nominal stress. At point *a* the residual stress τ_1 will be $\tau_y-\tau_1'$ where $\tau_1'=\tau_n d_1/d=\tau_n/\gamma$.

The residual stress τ_1 at *a* then becomes, using Equation 16,

$$\tau_1=\tau_y\left(1-\frac{C_1}{\gamma}\right) \quad \ldots\ldots(17)$$

Likewise the residual stress τ_2 at the outer fiber (*Fig.* 5) will be equal to

$$\tau_2=\tau_n-\tau_y-\tau_3 \quad \ldots\ldots(18)$$

The stress $\tau_y+\tau_3$ may be taken from Equation 13 using $\rho=d/2$. This gives, taking $\gamma=d/d_1$ as before,

$$\tau_y+\tau_3=\tau_y\left(1+\frac{\gamma-1}{k}\right) \quad \ldots\ldots(19)$$

Using Equations 16 and 19 in Equation 18 the residual stress τ_2 at the outer fiber becomes

$$\tau_2=\tau_y\left(C_1-1-\frac{\gamma-1}{k}\right) \quad \ldots\ldots(20)$$

As before these residual stresses must be added to the load stresses to find the resultant distribution. Referring to *Fig.* 6, if the nominal load stress at the outer fiber is equal to τ, the stress distribution represented by line *od* must be added to the residual stress distribution represented by the broken line *oec*. This gives a resultant stress distribution represented by the full line *oab*.

The resultant stress τ_4 at point *a* (*Fig.* 6) will be equal to the residual stress τ_1 plus the load stress τ_3. Since $\tau_3=\tau/\gamma$ as before,

$$\tau_4=\tau_1+\frac{\tau}{\gamma} \quad \ldots\ldots(21)$$

where τ_1 is given by Equation 17.

The resultant stress τ_0 at the outer fiber (point *b* *Fig.* 6) will likewise be

$$\tau_0=\tau-\tau_2 \quad \ldots\ldots(22)$$

where τ_2 is given by Equation 20.

The application of these equations will be illustrated in the following examples.

A hot-wound spring has a yield stress $\tau_y=100{,}000$ pounds per square inch in torsion. Also the ratio k between the initial slope of the stress-strain curve and the slope during strain hardening is equal to 4 which will give an approximation for some spring materials. If cold setting is carried out to the point where $\gamma=3$, i.e., the ratio $OE/OD=3$ for the load-deflection curve in *Fig.* 1,

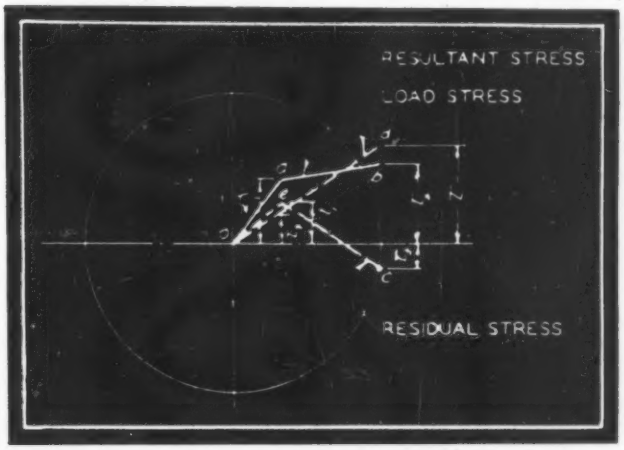


Fig. 6—Resultant stress distribution at load with strain hardening

then from Equation 15, $C_1=1.74$. From Equations 17 and 20 the residual stresses τ_1 and τ_2 at points *e* and *c* (*Fig.* 6) are as follows:

Residual stress at *e*:

$$\tau_1=\tau_y(1-C_1/\gamma)=100{,}000(1-1.74/3)=42{,}000 \text{ psi}$$

Residual stress at *c*:

$$\tau_2=\tau_y\left(C_1-1-\frac{\gamma-1}{k}\right)=$$

$$100{,}000(1.74-1-.5)=24{,}000 \text{ psi}$$

If a load P_y is placed on the spring, giving a nominal stress $\tau_y=100{,}000$ pounds per square inch, the resultant stresses at points *a* and *b* (*Fig.* 6) are

At point *a*, from Equation 21,

$$\tau_4=\tau_1+\tau_y/\gamma=42{,}000+33{,}000=75{,}000 \text{ psi}$$

At point *b*, from Equation 22

$$\tau_0=\tau-\tau_2=100{,}000-24{,}000=76{,}000 \text{ psi}$$

Since the maximum stress will occur either at point *a* or point *b* (*Fig.* 6) this means that after cold setting the peak stress will be 76,000 pounds per square inch as compared with a stress of 100,000 pounds per square inch for the same load before cold setting. This example indicates a reduction in peak stress at working loads equal to about 25 per cent due to the cold-setting operation.

As another example, a music-wire spring of .06-inch diameter is required for an ordnance application where a working stress of 170,000 pounds per square inch (neglecting curvature) is necessary because of limited space

(3). If the spring index is 6 the calculated peak stress including curvature will be around 210,000 pounds per square inch, but where static loading or light service is involved this peak stress may be neglected for design purposes (4). Such a spring would be made of material having an ultimate in tension of around 310,000 to 340,000 pounds per square inch in this wire size and a yield point around 85 per cent of this or 260,000 to 290,000 pounds per square inch (5). On the basis of the shear-energy theory, the yield point in shear should be around 58 per cent of this or 150,000 to 170,000 pounds per square inch. Assuming a value of 170,000 pounds per square inch, if the spring is cold set with conditions similar to those in the preceding example, the actual working stress at a nominal load stress of 170,000 (neglecting curvature) will be about .75(170,000) or about 130,000 pounds per square inch. Such a stress may be satisfactory in small-sized springs of this type for cases where limited life is required.

Is Even More Beneficial for Small Indexes

As mentioned previously, the above analysis applies strictly to large-index springs only but may be expected also to yield an approximation for moderate-index springs. Where small-index springs are involved the elastic strains near the inside of the coil will be much larger than those at other points around the outside of the section. This means that the residual stress at the inside of the coil will be considerably larger than the value of τ_2 (Equation 20) calculated on the basis of a large-index spring. At the same time, because of greater strains at the inside of the coil resulting in greater strain-hardening effects, the stress during cold setting at this point will be somewhat higher than for the corresponding large-index spring. As a result, the beneficial effects of cold setting are probably even more pronounced for low-index springs than for large-index springs.

Provides Estimate of Stresses

By using the methods given, the residual stresses set up as a consequence of cold-setting or scragging operations during the manufacture of helical springs may be estimated. This should throw some light on optimum amounts of cold setting for materials with different work-hardening properties. By superimposing nominal load stresses on the residual stresses, the resultant stresses at operating loads may be estimated, and the reduction in peak stress due to cold setting determined. The examples given show that this reduction may be considerable and explain why extremely high calculated stresses may be used in some cases in springs under light service or static loading.

Where fatigue loading for relatively few cycles of stress is involved, the effects of cold setting may also be beneficial—particularly if combined with shot-blasting operations. One reason for this is that by cold setting the mean stress of the stress range is reduced, and for some materials the endurance range thereby increased. This effect may be particularly pronounced for small-index springs where very high residual stresses may be induced at the inside of the coil with a correspondingly high reduction in mean stress at this point.

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3. *Manual on Design and Application of Helical and Spiral Springs for Ordnance* published by S.A.E. War Engineering Board.
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Reducing Causes of Chafing

CHAFING, galling, or fretting corrosion, as the phenomenon is variously called, is a problem which has become more important in aircraft-engine parts as the power output has increased and caused a corresponding increase in vibration problems. The phenomenon is characterized by apparent picking out and flow of metal on surfaces which are supposedly rigidly clamped one to the other. Picking out and flow of metal increases the stress concentration in the area where it is present which, of course, leads to reduced strength and, in many cases, failure. Since the phenomenon had an injurious effect on aircraft-engine parts in service operation, an investigation was initiated to determine conditions which would cause chafing and to discover, if possible, various means of eliminating it or neutralizing its effects.

From these tests and others, the following general conclusions can be drawn:

1. Chafing can be prevented by:
   a. Increasing the compressive load to a value where all sliding motion is eliminated.
   b. Providing a gasket which can absorb the motion or allow motion against steel without pickup.
   c. Providing a coating which can serve as a shear member or which can provide an antifriction surface. It is believed that those coatings which are effective in preventing chafing act as an antifriction bearing with many infinitely small balls or rollers. Lead plate is probably effective because of inherent lubricating properties in lead.
   d. Providing a plated or otherwise treated surface to increase friction greatly and to stop all sliding motion.
2. The severity of chafing:
   a. Increases with increase of motion at a fixed load up to the point where the motion applied will cause an oil-film wedge action to be maintained.
   b. Increases with increase of compressive load if a given sliding motion is maintained.
3. Surfaces of unlike metals in general chafe less than like metals.
4. Steel surfaces of the same finish chafe more than surfaces of different finish roughness.—*From a paper by H. C. Gray, Wright Aeronautical Corp., and R. W. Jenny, Curtiss-Wright Development division, at the recent S.A.E. National Aeronautic meeting in New York.*

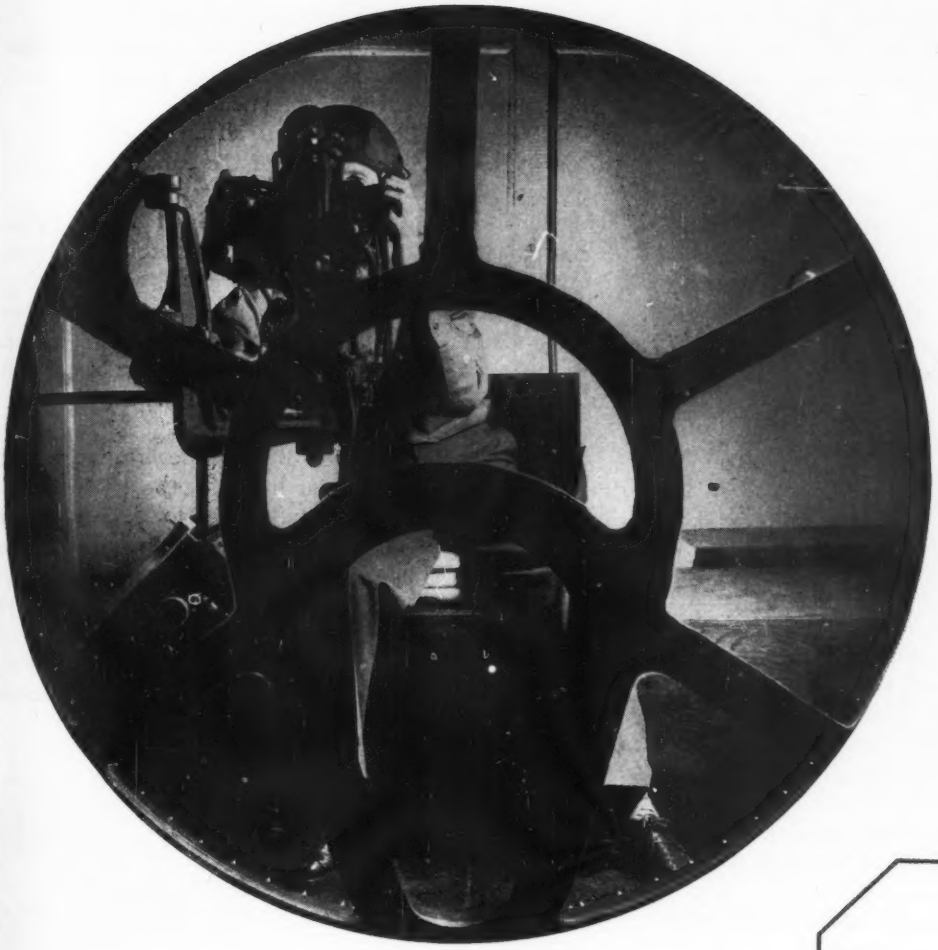
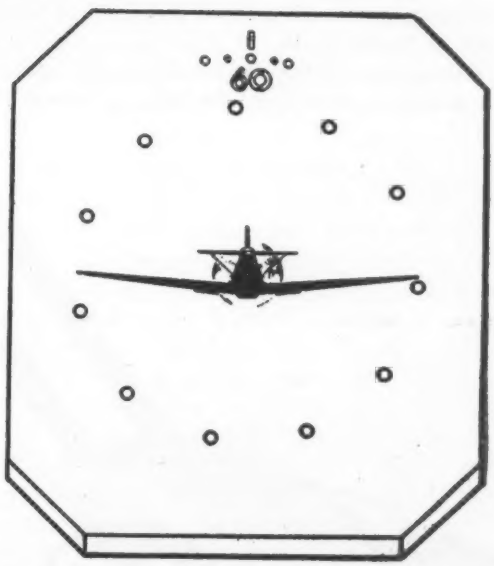


Fig. 1—One of the five remote-control stations on the B-29 Superfortress is located in the nose of the ship

Fig. 2—Below—This is what the B-29 gunner sees through his sight. Spot of light centered on nose of plane is aiming point



How the B-29 Concentrates Its Fire Power

By H. T. Hokanson and T. S. Lisberger
General Electric Co.

ALTHOUGH not our most heavily armed airplane, the B-29 Superfortress has been called the best gunned. Of the five turrets—two upper, two lower and one in the tail—as many as three can be controlled and fired simultaneously by a single gunner from a single sight, more than thirty different combinations of turrets being available to the five gunners.

Upper forward and lower forward turrets are placed ahead of the propellers. This gives them unrestricted freedom in fighting off attacks in the forward hemisphere, and against head-on nose attacks they team together for concentrated fire power. The upper rear turret provides protection from overhead attacks, the lower rear from attacks underneath. These two work together in stopping attacks from either side. The

tail turret, with its machine guns and cannon, stands guard against offenders from the rear. All of these turrets are directed by remote control from five sighting stations, *Fig.* 1.

Because of the extreme altitude, speed and distance performance of the B-29, previous armament methods were impossible. Hitherto, large formations of bombers have been used for mutual protection, but experience has shown that "throttle jockeying" by planes trying to maintain these large formations can use enough gasoline to cut their range substantially. Adequate fire power and fire coverage permit operation of small formations or single B-29's. In this way "throttle jockeying" is eliminated and longer range results.

Need for Remote Control Power Drive

It was obvious from the beginning in considering the requirements of the B-29 that it would have to make long flights to attack Japan. This meant that the gunners had to be kept comfortable and thus it was essential to locate their stations in the pressurized cabin. But the gunners cannot shoot guns through a pressure-tight wall, hence the guns must be kept on the outside while aimed and fired from inside the cabin. Furthermore, the large wind forces exerted against the barrels of machine guns protruding from high-speed aircraft made power drives for these guns imperative. Without this assistance the gunner could not instantly swing his gun on to a precise aim at a distant enemy fighter plane.

Indicative of the variety of attacks against which the armament system must be able to defend the plane instantly are the following typical situations:

1. Nose attacks in which only three seconds elapse between the time when the attacking fighter is nearly half a mile away until he is gone behind

2. Side attacks in which the relative angular rate is so great that the guns must lead the fighter by as much as 200 yards

3. Slowly closing tail attacks that give the fighter plane a long time to fire, thereby demanding extremely accurate return fire from the B-29 to win the duel.

Inasmuch as weight and drag are the worst enemies of high-altitude, long-range and large-bomb-load characteristics, it was essential to keep the total number of guns to a minimum. With remote gun-fire control the effectiveness of each gun is greatly increased.

In selecting the caliber of the guns several factors had to be considered. The guns had to be heavy enough to provide the hitting power necessary to destroy enemy fighter planes, yet it was also imperative to select the smallest caliber guns that would be lethal, to permit carrying the most ammunition for the allowable weight. More than two rounds of .50-caliber ammunition can be carried for the weight of one round of 20-millimeter ammunition. Hence the .50-caliber machine gun was selected for the job with one exception: The Boeing tail turret was to be provided not only with .50-caliber machine guns but also with a 20-millimeter cannon to give hard-hitting fire power against those slowly closing tail attacks. All other turrets would have two .50-caliber machine guns for the optimum design, each gun firing at the rate of 800 rounds per minute.

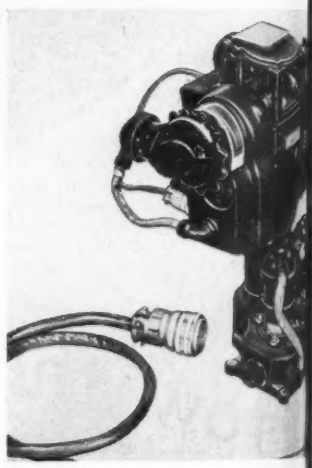


Fig. 3 — Left — Upper two-gun turret provides 360-degree rotation and elevation up to 90 degrees. Cover dome is removed to show the mechanism

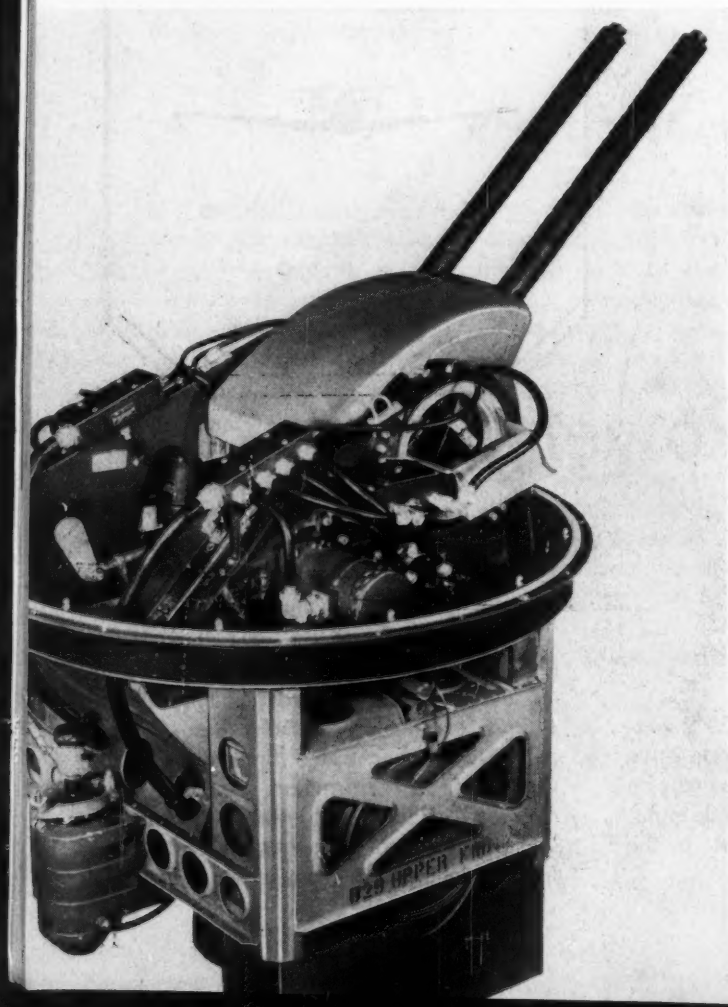


Fig. 4—Right—Sighting station contains slanting glass through which gunner sees target and sighting dots, Fig. 2

A good turret is the first essential of a good fire-control system. Careful attention was given to obtaining the maximum fire coverage from each gun in order that there be adequate protection without blind spots. Great freedom of motion is provided in each turret so that each gun can fire on both sides of the airplane, forward and aft as well as straight up for the upper turret and straight down for the lower turret. Each turret is mounted on bearings in such a way that it can swing around about a vertical axis. The guns themselves are fastened to a saddle in the turret that permits them to swing up or down, *Fig.* 3.

Ammunition for the upper and lower turrets is carried in ammunition cans which are a part of the turret. An-

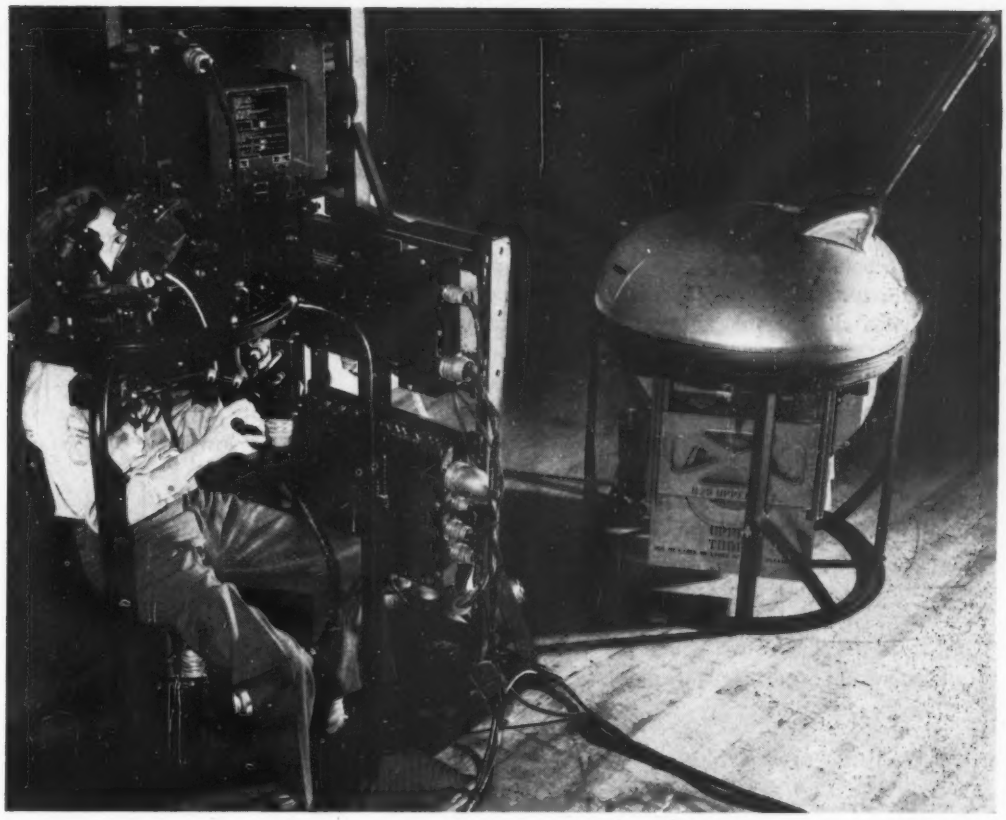


Fig. 5—Complete remote-control system includes sighting station, computer in background, and gun turret

tomatic ammunition boosters are installed for each gun. Each gun is provided with an automatic gun charger which performs two functions: It fires the gun by remote control whenever the trigger on the gunner's sight is pressed, and it throws out a bad bullet and puts in the next one whenever the gun finds one that won't fire. Six electric motors are used to operate each turret and point its guns as the gunner aims his sight.

A gunner concentrating on tracking a fast-moving enemy airplane might swing the line of fire from his guns through parts of his own airplane. Automatic electric fire interrupters stop the guns from firing at the gunner's own airplane and relieve him of this responsibility. At the same time these interrupters are accurate enough to permit him to fire within inches of parts of his plane, so that he loses the least possible fire coverage. An additional safeguard is the mechanical contour follower which prevents guns pointing at personnel in the plane. This is done because cartridges sometimes "cook off" in a hot barrel.

What the Gunner Sees

The gunner uses a sighting station to aim at and range the enemy fighter, *Figs.* 1 and 4. Mounted on the sighting station is the actual sight through which the gunner looks when he aims at the target. In this sight he sees a tiny spot of light which is his aiming point, *Fig.* 2. Whenever he puts this sighting dot on the enemy airplane he knows that his aim is correct.

Around the sighting dot is a circle of other tiny dots of light. The size of this circle can be changed by twisting the range control on the handles of the sighting station. When the gunner has his aiming point on the center of the enemy airplane, he adjusts the size of the circle until it just spans the length of the target. Sun or sky filters and a brightness control on the sighting light give the gunner a wide range of adjustment. With the proper settings, he can sight at a plane coming at him almost directly out of the sun or out of the blackness of night.

How the Sight Is Operated

The sight is in a small box open at both ends and containing a slanting glass, *Fig.* 4. When a gunner detects an enemy plane on the sight, he registers the size of the fighter on an instrument. Then he adjusts the circle of dots from tip to tip on the image of the enemy plane and keeps it there by moving the sight either to right, left, or up and down and by turning his range handle. From the size of the plane and of the luminous circle, the range is calculated by the computer. From the range and the movements of the sight to keep the enemy plane in the set circle of dots, the speed of the target plane is arrived at through a gyroscope functioning on the sighting device. The range and speed of the enemy are relayed by electrical impulses to the computer.

The sight is moved by two handles on which are incorporated trigger switches and action switches. When the gunner presses on the action switch, the turret that he controls is in operation. When he releases it the gunner who has secondary control of the particular turret can take it over.

Transmission of signals from the sighting station to the turret is effected by means of selsyn units, and an associated electronic servo amplifier relays the signal to the field of an amplidyne motor generator. This causes the amplidyne to generate a voltage which actuates the turret drive motor until the selsyns signal that the guns are lined up with the sight. The turret drive motor has sufficient power to lift a 200-pound man hanging on the end

of the gun. Each gunner is provided with the necessary switches and controls to start up and operate all of his armament right from his station.

From the foregoing discussion it will be evident that the selsyns, electronic servo amplifier, amplidyne, and turret drive motor cooperate to make the guns point wherever the sight points. Corrections for parallax, windage, gravity drop and lead are combined in a computer, visible in the background of *Fig.* 5, and superimposed on the sight direction signal, causing the guns to point so that the bullets will hit the target. The corrections are as follows:

Parallax is the error due to the fact that in the remote control system the gunner at his sight and the gun he is firing are a number of feet apart. Hence the aim of the guns must be angled toward the gunner's position, making up exactly for the distance between the sight and the gun at the point where the bullet meets the target. The computer makes this correction, which is highly variable, depending on range and angle of fire.

Windage tends to curve the course of a bullet rearward from the plane from which it is fired. Thus, a bullet fired broadside from a plane moving 250 miles an hour will be subjected to a gale of that velocity. With the range 800 yards at 3000 feet altitude, the wind will curve a bullet fired from a plane going 250 miles an hour as much as 35 yards. Where the air is less dense, as at 30,000 feet, the windage effect on the bullet would be only 12 yards, other conditions being the same.

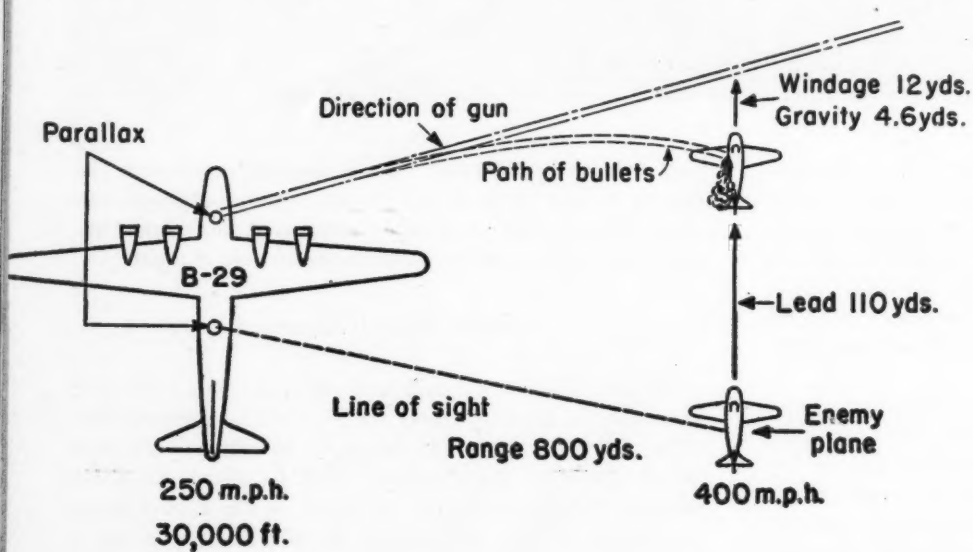


Fig. 6—Sketch shows how corrections calculated by the computer for parallax, windage, gravity, and lead are applied so that bullets will hit target

Ballistic Corrections

The amount of elevation needed to take care of gravity depends, of course, on range and altitude; the gravity correction is figured by the computer. With the range 800 yards, the .50-caliber bullet will drop 4.6 yards at 30,000 feet altitude. Windage and gravity are ballistic corrections which depend upon a knowledge of the altitude, outside temperature and speed of the plane. Since the navigator has the instruments showing these data, he sets dials to send them to the computer.

Lead is the correction for the relative movement of the plane and target. While a .50-caliber bullet is traveling 800 yards at 30,000 feet altitude, a fighter plane going 400 miles an hour at the same height will move forward 110 yards. Thus the guns have to be fired at the proper angle in front of the enemy plane to hit it. The computer determines the lead, making the guns point well ahead of where the gunner actually is focusing his sight.

Magnitude of Total Correction

If the foregoing corrections were not made and the gunner aimed directly at the enemy plane, the bullets would pass 122 yards behind the enemy plane and 4.6 yards lower than it, in addition to the parallax difference. *Fig.* 6.

Because all sight signals are transmitted electrically, more than one turret can readily be connected to these signals. Turrets can be transferred from one sight to another by simply operating a switch, usually at the direction of the chief gunner. In this way the fire power from all turrets can be distributed for overall protection or can be concentrated where needed the most. The nose gunner controls both the upper forward and lower forward turrets. The upper gunner controls the upper rear turret and can simultaneously control the upper forward when the nose gunner is not using it. The side gunners control the lower rear turret and can also control the lower forward when the nose gunner is not using it. The side gunners can also control the tail turret if the tail gunner is not using it.

Physical Strain Reduced

When using the B-29's remote-control system the gunners are emancipated from the cramped quarters and bitter exposure of the turrets used on other bombers, thus enabling them to maintain top efficiency due to the reduction of physical strain.

Another great improvement in the field of defensive fire made possible by this system is the relatively small number of guns which, due to their high degree of accuracy, are needed to protect unescorted B-29's from enemy fighters. This cuts down on armament and ammunition weight, permitting the Superfortresses to strike with loads up to 10 tons of bombs each on relatively short runs or to carry great gasoline loads, making possible record-breaking long-distance missions with lighter bomb loads.

An example of the effectiveness of this defensive fire from the relatively few guns on the B-29 was a recent four-hour battle between a lone Superfortress and 79 Japanese fighters during which seven of the enemy planes were shot down and the AAF bomber came through unscathed. An AAF officer aboard the B-29 reported that "the gunnery system functioned perfectly and the gunners coordinate their fire so well that they put a metal fence around the besieged Superfortress."

Specifying Rational Tolerances for Interchangeability, Low Cost

By F. A. Wedberg
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"BUMP to fit and file to suit on assembly," or words to that general effect were more or less common on aircraft drawings of just a few years ago. The final assembly area resounded with the ring of hammers and the screech of files on tortured metal. In those days, when thirty airplanes were a large production order, and they didn't stray far from their home base, this practice at least bordered on acceptability. Even now the educated hammer is still a useful albeit a much overworked tool.

However, the explosion of our healthy infant aircraft industry into sudden maturity under the pressure of the war effort has accomplished much in the development of tooling and production methods which has largely eliminated the necessity for the use of hammers, files, and crowbars on assembly lines. Our main concern—looking ahead—is not whether we can build aircraft in quantity—but how we are going to build them at a cost which will match the peacetime pocketbook.

Tolerances in Design

The solution of this problem must start with proper basic design and in that design, the use of rational tolerances which will permit economical tooling and production, eliminate handwork in the final assembly stage, grounded airplanes and lost profits when replacements become necessary in the field. Among the basic rules involved in the establishment of practical rational tolerances are:

1. Nothing in this world can be "right on the button," hence the need for the designer to establish tolerances, large or small, within which the tooling and production organizations may produce, and the inspection organization may accept, a part or assembly.

2. Tolerances on an assembly are equal to the sum of the tolerances of the individual component parts and hence may be equal to the sum of all the adverse tolerances of these parts. This is especially important where fits of various components or clearances of moving parts are involved.

3. It is essential that important physical points or faces—preferably *not* centerlines—be chosen as datum points or lines from which actual measurements can be made and upon which production tooling may be

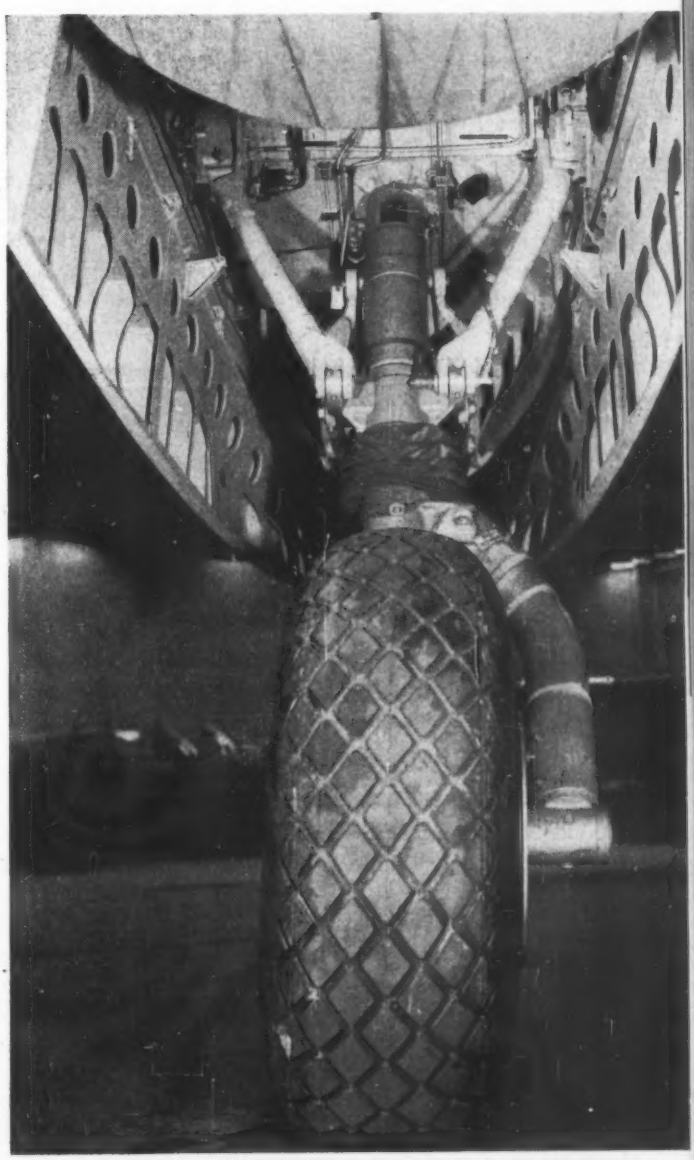


Fig. 1—Landing gear of C-46 airplane. Trouble-free mass production of such assemblies demands that tolerances for all parts be determined and specified competently

based. Such points, lines or faces should be chosen because of their basic importance and because of being common to, or directly affecting the mating of, the component parts.

4. Tolerances must be a compromise between acceptable deviations on the finished product and the cost of production.

5. Since cost is almost always critical, it may be stated in general that the allowable tolerances on detail parts should be the maximum consistent with engineering and quality considerations, even though in some cases it may be desirable to tool to much closer limits.

6. Tolerances should be based on the degree of accuracy obtainable with average, or somewhat below average, personnel and equipment in order that rejections and therefore ultimate cost may be held to a minimum.

7. The use of offset dimensions for locating points on a part will permit holding much closer tolerances than when such locations are given by the use of angles.

8. Effect of temperature on dimensions where dissimilar materials are used must be recognized, both in the case of close-fitting machined parts and in large assemblies where dimensions between important points approximate 100 inches or more. In considering temperature effects, it is important to include not only the materials used in the assembly but also the methods and materials used in the tools for producing the parts. For example, the use of steel fixtures for producing large aluminum alloy assemblies should be avoided, as errors on the order of .00025 inch per inch of length may result under conditions where temperatures vary as much as 40 degrees Fahr. Such variations are entirely possible, particularly under conditions where components are fabricated at

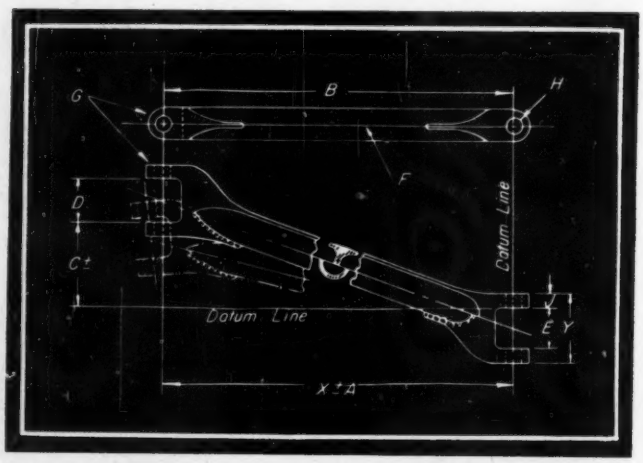


Fig. 2—Typical forged or welded arm has datum lines through hinge holes and at machined surface of right lug

widely separated geographical locations, having corresponding differences in climatic conditions. Operating temperatures in fighting aircraft range from −65 to 150 degrees Fahr.

9. Tolerances in many cases should vary in geometrical proportion to the size of the parts. For example, deviations in airfoil contours or clearances between moving airfoil components on a small, fast fighter must be held much closer than corresponding dimensions on a large bomber or cargo plane, because of aerodynamic considerations as well as possible tooling inaccuracy and variations of the type referred to in Item 8.

10. The tool designer must, in laying out his tools and fixtures, use only a fraction of the total drawing tolerance in order that the finished product may, with the further deviations resulting from tool wear, part and unit distortion, and other production factors, fall safely within the established drawing tolerances.

11. In the interests of holding rejections to a minimum, with resulting economy, it is useful to retain a tolerance reserve which probably will not be indicated on the production drawing, being available only for the use of inspection or salvage personnel. This reserve may be used to cover unpredictable variations which result from riveting, welding or heat treating. It may also include allowable minor springing of more or less flexible units such as control surface hinges during assembly, and may constitute purely temporary increased limits, allowed only during early production runs. Recognition of the usefulness of this practice is evidenced by the recent growth of the Army Air Forces Materials Review Procedure which utilizes these reserve tolerances.

Many Phases of Engineering Involved

SCOPE: Establishment of tolerances should include not only details, subassemblies and assemblies, but also acceptable limits in bench testing of accessories, installation and operating tolerances in final assembly, and acceptable performance ranges for ground and flight testing. Another field which is receiving increasing attention is the establishment of more scientific methods and equipment for measurement of surface roughness, incorporating the use of standard roughness sample blocks supported by precision machines such as the Brush Analyzer and the Profilometer.

TYPES OF TOLERANCES: There are three general tolerance classifications: Unilateral, bilateral and range. The unilateral type, with limits extending in one direction only, are desirable on cylinders or holes, for example, 1.500 plus .015 minus 0, or 1½ plus 1/64 minus 0. In the case of their mating parts, such as pistons or bolts, it is desirable to establish tolerances of plus 0 minus a decimal or fraction.

Bilateral tolerances are those having limits in both directions. For example, 1.500 plus or minus .015 or 1½ plus or minus 1/64. This type of tolerance is desirable on such items as distances between hole centers, or on welded or riveted assemblies where variations cannot be controlled in one direction.

Range tolerances are those where minimum and maximum values only are given. For example, 2000-2050 rpm or 75-80 lb. This method of specifying tolerances is used by many concerns on machined parts. It is particularly adaptable for indicating the scale of permissible variations for operating mechanisms, including revolutions per minute, temperature, pressure, vacuum, times, torques, flows, and similar instrument readings, thus eliminating the necessity for mechanics, inspectors, pilots, and other personnel to go through mental gymnastics in order to pass on the acceptability of observed readings.

SHEET METAL DETAILS: As noted in the foregoing, it is desirable wherever possible to pick a definite point, line or face as the basic datum for all dimensions. In some cases, however, such as curved bulkhead sections,

or wing ribs, it is found useful to create datum points by the addition of form holes which are used as the basis for all tooling, fabrication and assembly operations.

An important problem for the engineer, which may well have a great effect on the cost of the product, is that of conveying to tool design and production personnel the relative importance of various dimensions. It is customary in most engineering organizations to indicate, in the drawing title block, general tolerances which apply in sheet metal fabrication or machining processes for all dimensions not otherwise qualified by individual tolerances. Such general tolerances commonly used are plus or minus .015 or .030 for sheet metal and .010 for machining operations.

It has been common practice among shop and inspection personnel to temper the use of such tolerances with individual judgment in recognition of the known fact that many dimensions could exceed these general tolerances without jeopardizing the acceptability or efficiency of the piece or assembly in question. This practice has led to some embarrassment during our current production emergency due to the high utilization of inexperienced personnel in all phases of engineering and production, indicating the need for more detailed expression of tolerances on production drawings.

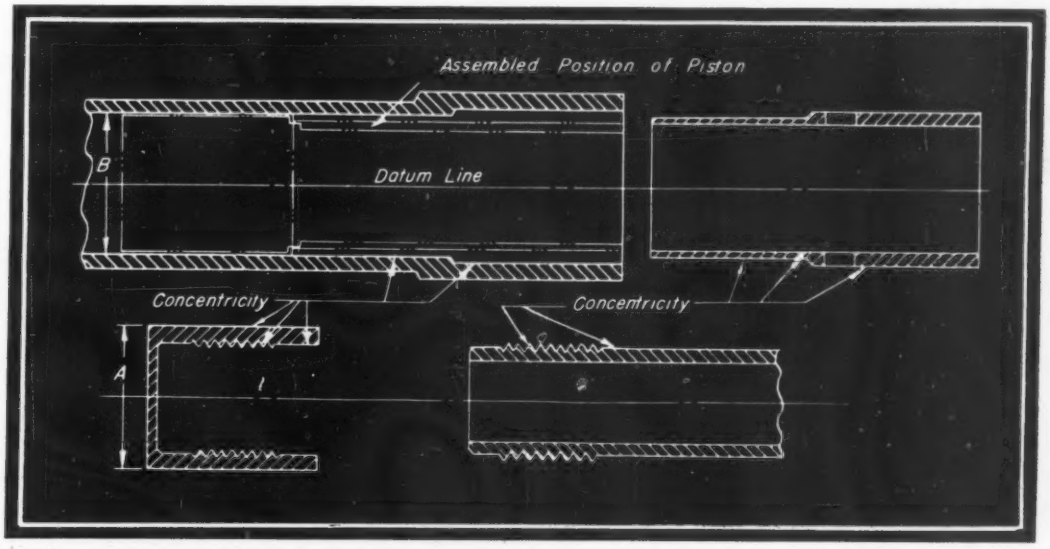


Fig. 3 — Tubular machined-part assembly requires tolerances on concentricity as well as on basic diameters

Closer Collaboration Required

Necessity has been found for closer cooperation among design, production and inspection personnel because in many cases the people who have been engaged in actual production and inspection operations have a better knowledge of the practical tolerances which may be held and which have been passed as acceptable under the regime where individual judgment was more freely used than is practicable under present conditions.

One method which has been used to indicate general tolerances is that of specifying dimensions to one, two, three, and four decimal places, each decimal place representing an established degree of accuracy. Another similar procedure uses decimals for dimensions requiring relatively close tolerances and fractional dimensions for those which are less important. Such practice permits the tooling and production organizations to take advantage of less costly methods.

FORGINGS AND CASTINGS: It often is useful to provide bosses or other flat areas which permit the drilling of basic holes, somewhat similar to those described under Sheet Metal Details, from which the important machining dimensions and faces may be located. These in turn become the datum lines for the balance of the dimensioning system. This practice saves much detailed layout time and production scrap of complicated units which otherwise present difficult problems to the tool designer and the machine operator. By establishing such tooling points and an improved tolerance base as described in the following it is possible to hold forging dimensions to insure minimum machining. Further economy can be effected by coin-pressing local bosses to close tolerances, thus eliminating the necessity for machining.

There is a definite need for standardization of acceptable tolerances for castings and forgings among aircraft manufacturers working in cooperation with the vendors who produce these materials. Such standardization would insure more uniform products and give the vendors the obvious advantage of working to the same tolerances for all of their customers. Some studies are now under way and it appears entirely possible to set up such tolerance standards on a simple base. As an example, in the case of sand castings an acceptable basic tolerance of plus or minus 1/32-inch for dimensions up to four inches, with an increased tolerance of plus or minus 1/64-inch for each additional two inches, is considered reasonable. It further is believed that the basic differences between aluminum and magnesium alloys, steel, brass, bronze and cast iron are so slight that the same base tolerances may be used for all of these materials. The above basic tolerances apply where dimensions lie wholly within a single portion of the die and need to be increased somewhat for dimensions affected by core location or which cross the die parting lines and are hence affected by the degree of closure or mis-mating of the patterns or dies. It also is essential that tolerances be set up to express the acceptable degree of deviation from flatness or straightness in terms of unit of length.

MACHINED PARTS: The wise use of tolerances on parts

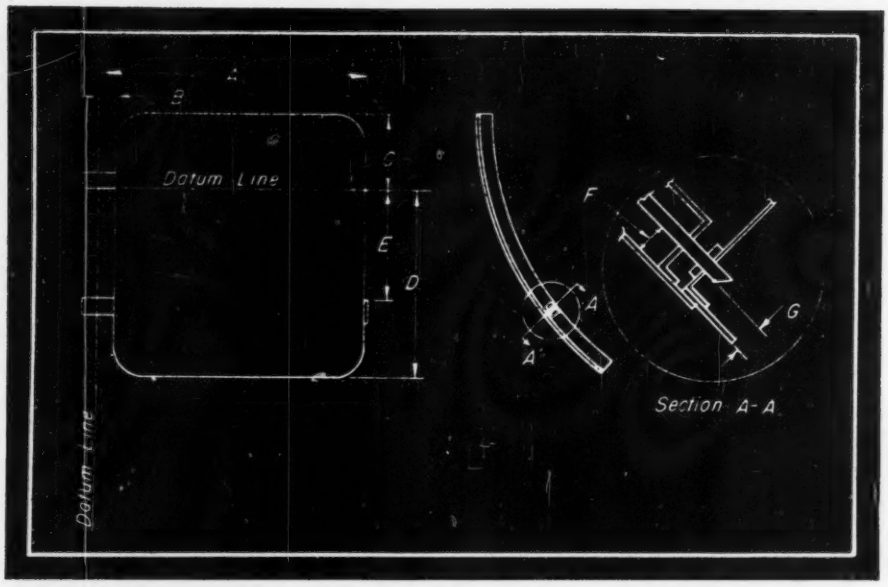
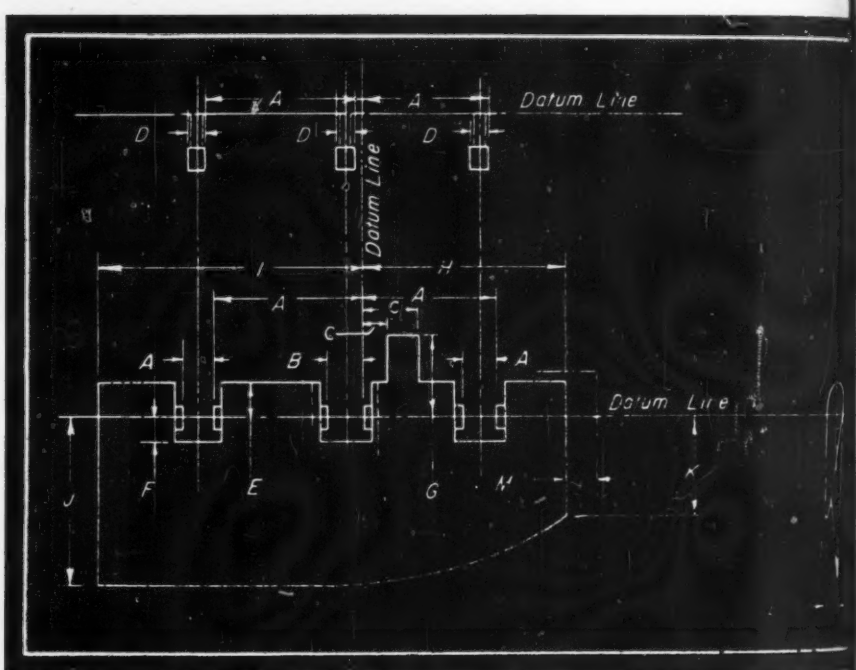


Fig. 4—Left—Typical aircraft door. Datum lines are established at the hinges which are common to the opening in the body and the door

Fig. 5—Below—Detail of a control-surface member. Here again datum lines are located through hinge holes and at hinge surface



in this category can do much toward keeping costs and headaches down. The comments on specification of general tolerances given under Sheet Metal Details are even more important in connection with machined parts due to the more costly nature of the process and the correspondingly greater monetary loss when parts are scrapped.

Shown in *Fig.* 2 is a typical welded or forged member. The following detailed comments are offered in connection with the dimensions noted: Datum Lines: All dimensions are given from the datum lines through the hinge holes at the right and one of the important faces of the right lug, these being the lines which determine satisfactory functioning of the part.

A: This tolerance should be as liberal as design considerations permit in order to hold rejections due to welding and heat treating distortion or mechanical errors to a minimum. The tool designer may find it practicable and desirable to tool this dimension to much closer limits. Common drawing tolerances for assemblies of this kind are plus or minus .002 or greater depending upon the importance of this dimension of the functioning of the part.

B: It is important in a case like this to indicate the degree of parallelism of the axes of the two holes to insure against the effect indicated by the dotted lines at the left side of the figure. A typical note for this application is "centerlines of bolt axes to be parallel within five minutes," the exact figure depending on the length of the member and the clearance between the bolts and holes.

C: It usually is necessary to hold this dimension to relatively close limits to insure proper assembly with mating parts. The value of this tolerance depends on the lateral clearance between the mating parts and between the end bolts and their holes which in turn will be determined by structural considerations. For example, there would be no justification for holding the *C* dimension to plus or minus .001 if there was lateral clearance between the bolts and holes at the end connections which allowed a displacement of plus or minus .010. In a typical case of this type, there is probably a minimum clearance between the bolts and holes of at least .003 which, with a ratio of *X* to *Y* equal to ten would be equivalent to a tolerance on the *C* dimension of plus or minus .030. Attention is further directed to the fact that with *Y* equal to five inches the .003 bolt to hole clearance would be equivalent to limits on parallelism of the bolt holes of about two minutes per end, or four minutes total.

D and *E:* Tolerances dictated by structural considerations such as extent of reverse loading along the bolt axis. It is customary to choose the basic dimension from standard milling cutter widths with tolerances adjusted to allow for errors due to run out and wear of the cutter.

F: It may be found desirable from a structural standpoint to indicate the acceptable tolerance on straightness of the tube or forging to cover variations in the basic unit

as well as those resulting from welding or heat treating strains. The probability of off-center location of the end holes with respect to the normal centerline of the member should also be considered.

G: Another item of importance in structures of this type is to indicate the minimum thickness and width of material around the end bolt holes for structural reasons and it may also be desirable, if clearances are close, to indicate the maximum acceptable dimensions.

H: In highly stressed parts with reversible loading it is sometimes desirable to use special close-tolerance bolts and holes to match, with corresponding increase in cost.

Tolerances in a Machine-Part Assembly

In *Fig.* 3 is shown a typical machined-part assembly such as is used on a hydraulic strut in which there is a piston operating in a cylinder, the packing gland being omitted as irrelevant to this discussion. A study of this unit indicates the obvious necessity of establishing tolerances not only on the basic diameters, but also on the concentricity of these diameters with respect to each other in order to insure free operation of the assembly. The specification of roughness tolerances is useful in applications of this type where there is contact between moving parts.

Another application where the proper specification of tolerances is important is that of doors such as the one indicated in *Fig.* 4, which is typical for landing gear, tailwheel or bomb-bay closure or for access to cargo and passenger compartments. It is obvious in this case that the datum lines must be based at the hinges which are common to both the main body opening and the door, and if proper fit is to be obtained all dimensions must originate from the datum lines as indicated. An item which is sometimes overlooked on the drafting board is that the latch location should be given with respect to the contour, rather than to some theoretical centerline which the door knows nothing about, because the relationship of the door and frame contours determines the proper mating of the latch.

It should be recognized also that due to the accumulation of manufacturing errors, it may be found desirable in assemblies of this type to allow excess stock around the periphery of the door for trimming at assembly. However, the main body framework should be held within the basic tolerances to insure interchangeability.

Control surfaces have provided many headaches for production personnel due to the necessity, despite their light construction, for maintaining proper hinge alignment, end clearance and conformity to proper contour. Where cloth covered construction is used, the strains resulting from shrinkage of the cloth caused by dope application result in distortions sometimes difficult to control. Assemblies using metal covering have a tendency, due to the thin gage of material, to wrinkle under the slightest distortions caused by riveting or assembly operations. Here again it is obvious that the hinges are common to both mating parts and therefore are used for the datum lines. It will be noted from *Fig.* 4 that the lateral datum line is chosen as a hinge face near the centerline of the assembly to keep the size of the long dimensions to a minimum.

Detail tolerances that have been found satisfactory in applications involving dimensions of fifty inches or less are as follows, the letters used being those of *Fig.* 5.

A: Plus or minus .020. It is difficult to hold much closer tolerances due to accumulated manufacturing errors such as those due to riveting.

B: Plus or minus .015 represents about the minimum which can be held unless shims or other means of adjustment are employed. It is desirable to hold this, or one of the other hinges, to close dimensions in order to transmit the lateral structural load.

C: It is desirable to provide increased clearance at these hinge points to absorb the accumulation of production and temperature variations. This is also acceptable structurally within reasonable limits since only radial loads will be taken at these hinges. This extra gap presents a problem if the hinges are provided with ball bearings whose inner races may need to be clamped together. It may be necessary to provide for adjustment by shimming or other method unless the mating hinge fittings can be sprung sufficiently to compensate for the above noted deviations.

D: Hinges, being machined surfaces, may be held to relatively close tolerances on the order of plus 0 minus .005 or .010-inch.

E and *G:* The plus tolerances in these cases should preferably be 0 but due to the riveted and formed-metal construction, tolerances of plus .030 minus .060 have been found reasonable for these applications.

F: The tolerance problem at this point is similar to *E* and *G*, but since the bottom of the slot usually is a free edge a minus 0 tolerance may be maintained and plus .060 has been found satisfactory.

H and *I:* Here again a plus 0 tolerance is desirable but difficult to hold and therefore a plus tolerance of .030 is indicated. The minus tolerance should be a function of the basic dimension and a figure of about two tenths per cent (1/10-inch for 50-inch) has been found satisfactory.

J and *K:* In designs using metal framework and covering, a tolerance of plus or minus one per cent of the basic dimension has been found reasonable. An additional allowance of about one-eighth inch must be provided for cloth covered units to include the build-up of tape, dope and sewing at the corners.

L: Surfaces of this type are subject to warpage due to riveting strains and/or the pull of doped cloth. Toler-

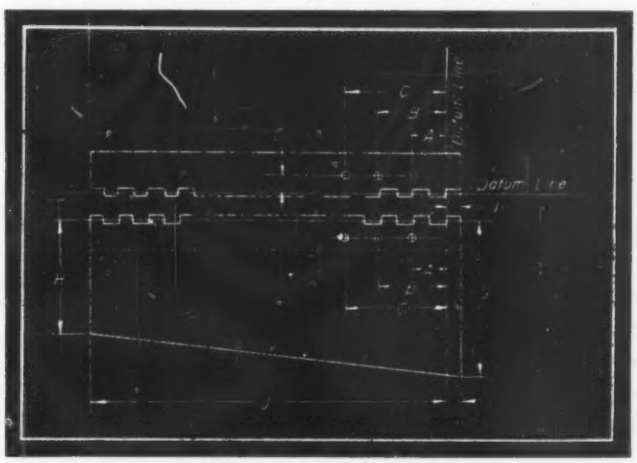


Fig. 6—Tolerances specified for a control-surface tab such as this include limits permissible for deviations from straightness of the hinge and the trailing edge

ances should be based on the span of the surface, a reasonable allowance being plus or minus three-tenths per cent.

M: The nominal clearance dimension at the ends of control surfaces should be at least one-quarter inch for metal-to-metal clearance and three-eighths inch where cloth-covered surfaces are involved. Tolerances should be the sum of those on the *H* dimension and the corresponding figures for the wing panel.

On comparatively flexible surfaces of this type it is necessary to allow certain deviations for inspection purposes (materials review tolerances). For example, a surface of this type, if supported at its two end hinges, will sag at the center hinge due to its own weight. Tolerances therefore should be set up covering the allowable deflection in assembling the unit into the checking fixture. Some allowance customarily is made for lateral springing of the hinges on the panel. These figures can best be set up through a study of each design by engineers and inspectors based on the experience of the first few production units.

Factors Applying to Control-Surface Tabs

Control-surface tabs of the type indicated in *Fig.* 6 using all-metal construction have been successfully produced with tolerances on the chord dimensions (*G* and *H*) of plus or minus one per cent with a minimum figure of plus or minus .050. On the span dimension *J* a value of plus 0 minus one-tenth per cent with a minimum of plus 0 minus .030 has been found satisfactory. The plus 0 limit can be maintained in this case due to the fact that the ends usually are free metal edges with allowance for final trimming as required. On surfaces of this type, it also is necessary to set up tolerances covering acceptable deviations from straightness of the hinge and the trailing edge in a direction perpendicular to the plane of the surface. Gradual deviations of .030 to .050-inch in ten inches have been found satisfactory. Chord section deviations from contour have been held to about three-tenths per cent of the chord with a minimum value of .030-inch.

Indicated in *Fig.* 7 is a typical dimensioning method for tying in the wing, its tip, and the aileron, using the aileron hinges and tip attaching holes as points common to each pair of units. Tolerances on the *A* dimension can be held to plus or minus .030, on *B* the same as discussed for *Fig.* 5, on *C* similar to *B* but with the plus and minus reversed, and values for *D* are comparable with those indicated for *M* of *Fig.* 5. It is essential that the tip attaching dimensions (*F* to *K*) all originate from one point to avoid accumulated tolerances. A customary tolerance on these dimensions is plus or minus .002. The type of construction involved makes it difficult to maintain interchangeability unless oversize holes are provided. Recent structural tests on a vibrating machine covering a medium-size tip have indicated that 1/16-inch oversize holes are entirely satisfactory for this application.

FINAL ASSEMBLY: The foregoing has indicated a small percentage of the tolerances involving deviations, clearances, etc., which are required by tooling, manufacturing and inspection groups in order that they may satisfactorily perform their functions. These data must be supplemented by much additional information regarding acceptable clearances and misalignments. Another field which must be covered is that of acceptable ranges including pressures in hydraulic and oxygen systems, torques of nuts and fluid-line connections, cable tensions, etc.

RAMP AND FLIGHT TESTING: A great number of tolerances must be provided to cover ground and flight testing. Among these are times for retracting and extending landing gears, wing and cowl flaps, revolutions per minute of power plant, temperatures, pressures and vacuums, and the other numerous values which must be checked in order to insure the proper functioning of the various installations in modern, complex airplanes.

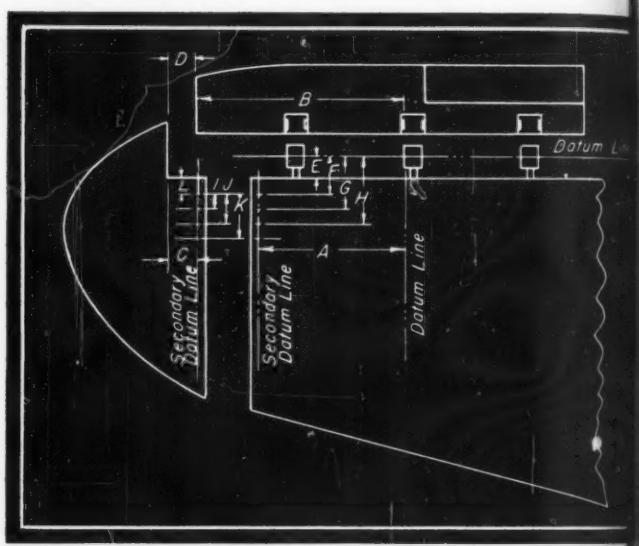


Fig. 7—This method of dimensioning properly ties in the wi , its tip and aileron. Datum lines for each pair of units are at points or surfaces common to both units

The great multiplicity of parts and mechanisms, used on aircraft of today many of which are very costly and must be handled with extreme care to insure their proper functioning, make it impractical to expect any individual or group to tool, produce, inspect, or flight test the various components or the complete airplane without extensive written information including proper tolerances. These data can be accumulated only through the diligent efforts and wholehearted cooperation of all groups involved. The success of the airplane from both a cost and performance standpoint, will be in direct proportion to the amount of effort and degree of cooperation of the team which produces it.

AIRCRAFT GAS-TURBINE engines for jet-propulsion planes require heating rather than cooling for lubricating oil. The reason, according to General Electric, is that there is only one moving part in the unit and, because that rotates without appreciable vibration, it can be supported on antifriction bearings. The oil heating system utilizes thermostatic control to maintain proper temperature under varying degrees of cold at high altitudes. Oil consumption is negligible, being less than 15 per cent of that required in other types of aircraft power plants.

Precision Casting— Its Place In Design

By S. O. Fisher and E. E. George
Schenectady Works Laboratory
General Electric Co.

LIKE most other processes used for making machine parts, the "lost wax" technique of producing precision castings is ideally suited to some types of parts and utterly unsuitable for others. It should not be looked upon as a cure-all—a panacea—for all the ills of other processes. Properly used, it will produce intricate parts to high accuracy in size and shape at a cost in line with engineering demands. Improperly used—specified where it is not actually required—it is much too costly. Thus, unless the designer works in close collaboration with producer, results are apt to be disappointing.

Just what *is* "precision casting"? As it stands today, precision casting utilizes a master pattern, made to include the overall net shrinkage of the process, to produce a cast mold of lead or a low melting-point alloy. This mold is gated, polished and made ready for filling with wax. Then wax patterns are made by casting under a vacuum, injection casting, or centrifugal casting. Next, the wax patterns are assembled to make a mount which becomes the final pattern.

After receiving a ceramic coating, the mount is placed inside a thin metal cylinder which forms the flask and is invested, the material generally used for investment being an air-setting silica mixture. After drying, the wax is melted from within and the mold brought to temperature for casting, usually from 900 to 1050 degrees Cent. Casting is done by placing the mold in a small arc furnace in which has been melted just sufficient

Fig. 1 — Supercharger buckets precision cast in a group. Sand-blasted surface finish on rough castings measures about 120 to 170 micro-inches

metal to fill the mold, and inverting the combination so that the metal will flow into the cavity. After this, the casting is cleaned and cut apart into the individual pieces desired and these are ground and cleaned to suit the application.

Precision castings as engineering parts have been confined largely to high-alloy materials. The reasons for this are fairly apparent, i.e., high-alloy materials generally are difficult to forge, fabricate and machine. Precision casting eliminates these three operations in many instances, or at least requires only a minimum of machining such as a shaving operation or finish-grind where extremely close tolerances are required. It seems unlikely that precision castings from low-alloy steels will ever compete economically with other methods of manufacture except in those cases where configuration of the part make conventional manufacturing methods excessively difficult or commercially impracticable.

Up to the present time, the largest application for precision-cast parts has been for the exhaust-driven supercharger for high-altitude aircraft. Also, several designs of permanent magnets for electrical apparatus are made by this method. In addition, there are designs of other applications which unfortunately cannot be mentioned at this dime.

Comparative costs between precision castings and other methods of manufacture depend on the requirements of the part and will be reflected by the ability of the engineer to coordinate his design with manufacturing practice. Certainly where a piece can be turned out for example on a screw machine, precision casting should not be considered. It is well to understand that today, as indicated in the fore-

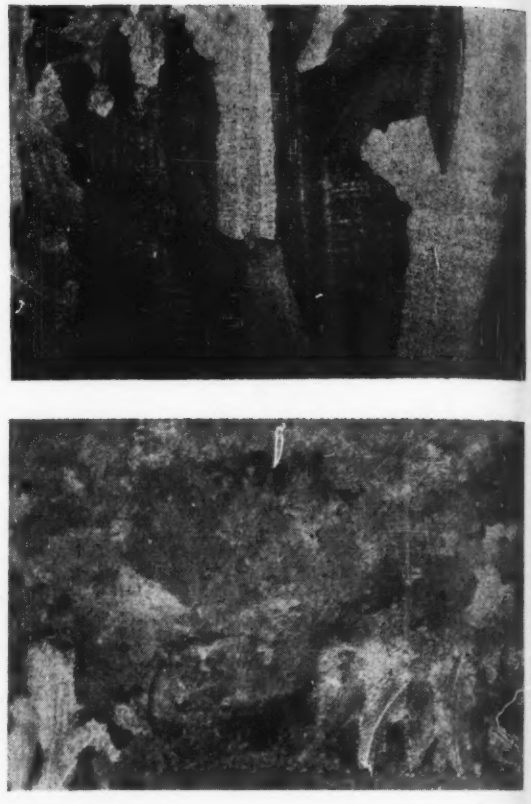


Fig. 2—Above—Macrostructure of precision-cast stainless steels. Top—Type 310 (25% Cr, 20% Ni). Lower—Type 410 (12% Cr)

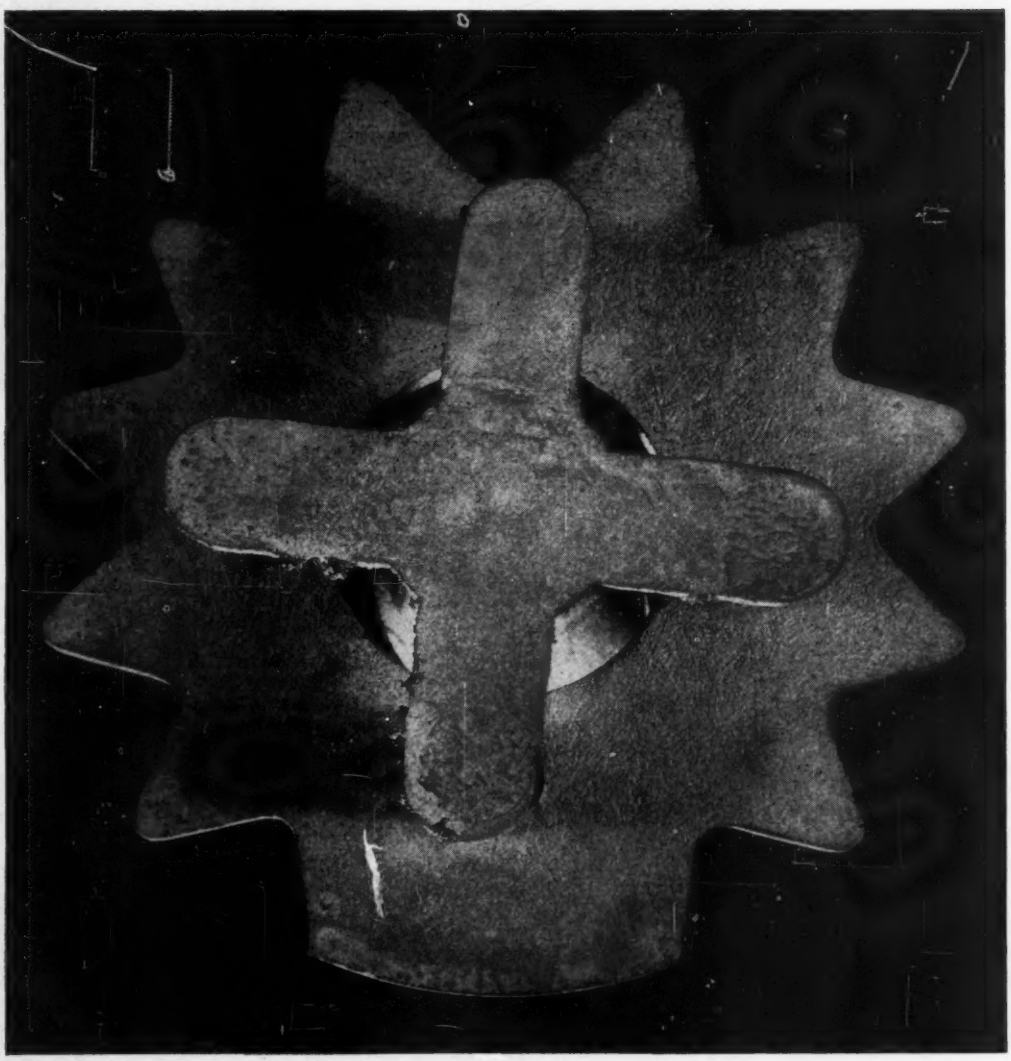


Fig. 3 — Left — Star wheel part of a cam switch precision cast from alloy containing 88% copper, 8% tin and 4% zinc

going, the process involves a large number of operations. Therefore, it becomes evident that if the piece can be made by another method equally as well and in fewer hours, precision casting should not be used.

As mentioned above, the oldest commercial application for this process outside of the dental and surgical field is the buckets on an exhaust supercharger. (see *Fig.* 1). When these first were produced they were only slightly modified from the original machined pieces. However, as experience was gained in precision casting, the engineers soon learned that they no longer had to consider the machinability of the contours involved. As a result, buckets now are being designed and manufactured with contours which can only be cast and with much improved efficiencies. Aerodynamic surfaces with intricate twists that rival the corkscrew for lack of straightness, are becoming the order of the day. Yet they fit in admirably with precision casting technique.

Possibilities of multiple-piece patterns seem limitless. Several castings are being produced where the wax patterns are made up of some 20 to 30 individual parts. This introduces a casting field where internal projections can be made without a mold involving numerous loose pieces.

Design Changes Readily Made

The cost of a mold with which to start the casting once the master pattern has been obtained is quite low, usually requiring only two or three hours to produce. If an engineer wishes to make slight alterations to a casting, he need only alter the lead mold and proceed immediately to make more castings. Thus, many combinations or alterations can be tried in a single day. When compared with a punch press or a screw-machine setup, the process is particularly adaptable to development work of this nature. Many methods of gating and risering also can be tried conveniently with only one pattern involved, making it possible for the manufacturer to determine acceptable castings from the first trials and still be able to make design changes at the same time.

Since uniform shrinkage is not always found in this process, it is always best to consult the manufacturer before making pattern drawings because each producer has his own shrinkage rules to follow. In addition, it is well to remember that master patterns cannot be promiscuously interchanged between producers.

In connection with shrinkage, it is not *metal* shrink that is important but the over-all *process* shrink. This involves such things as master-mold shrink, wax shrink, mold expansion in preheating, and finally the metal shrink itself. This explains why the shrink figure sometimes referred to is different from those used in connection with other casting methods.

The term "precision casting" as applied here may be misleading in these days of superfinishes and machine work measured to tenths of thousandths. Tolerances of plus or minus .003-inch to .004-inch appear to be as accurate as is practical for production precision casting at this date. Closer toleranaces have been obtained, but these are the exceptions rather than the rule. Tolerances will vary from this .003 to .004-inch range on dimensions below ¼-inch up to plus or minus .01-inch on 1-inch dimensions. The particular dimension in question should be discussed with the producer, because two equal dimensions will not always result in equal variations in the finished product. This fact is tied up with the same reasoning behind the statement previously made that shrinkage allowances are nonuniform.

Due to the fact that precision casting is effected in hot molds, internal shrinkage and gating problems are not the same as those

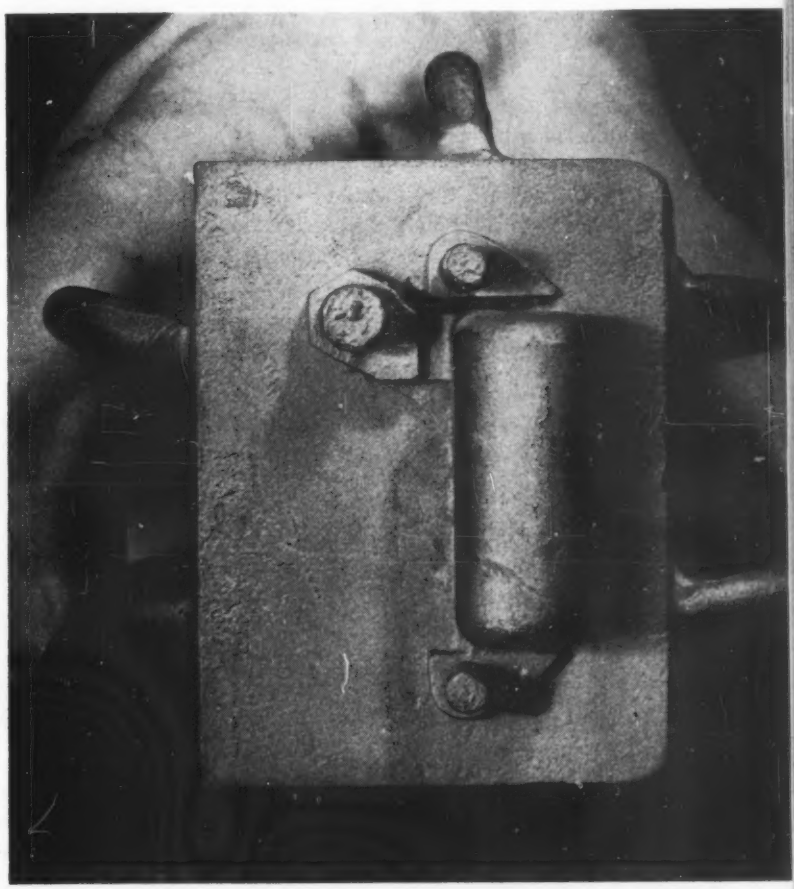


Fig. 4—Positive half of die for molding plastics. Such parts are precision cast to considerable advantage. Made of alloy steel they are held to good accuracy in shape and size

occurring in ordinary foundry practice. Large surfaces, such as would be found in trying to cast a 4-inch cube, have a tendency to collapse on the surface instead of shrinking internally. It therefore becomes a problem to hold outside dimensions on a piece of this type.

Generous fillets are just as desirable and drastic changes in cross-sectional areas as undesirable in this process as in other casting techniques. It has been found from experience that a part which ordinarily would be designed with several different cross sections in order to make casting possible by the usual methods can be redesigned into uniform cross sections and still be cast by the lost-wax process. The result is a better part yet one which can be readily manufactured. This gives the engineer an opportunity to combine increased efficiency with the kind of sections a

foundryman finds most desirable. Ornamental design on a casting often can be included in precision castings where it would be impracticable with sand-mold castings. Most of the above advantages are due to the fact that the pattern is of a destructible nature and not one that has to be removed intact for subsequent casting of additional parts.

As has been indicated, precision casting consists of casting molten metal into a refractory mold which has been preheated to some temperature well into the red color range. This high degree of preheat is necessary to assure complete filling of the mold. The mold is a poor conductor of heat, and this together with preheating results in a slow rate of cooling during the solidification of the metal. Typical examples of the macrostructure resulting from this slow rate of cooling are shown in the photographs of *Fig.* 2. Physical properties of the same materials (stainless steels Types 310 and 410) as revealed by tests conducted on sets of three specimens are listed in TABLES I and II.

TABLE I

Properties of Type 310 Stainless (25% Cr, 20% Ni) As Cast

Test Piece	Tensile Strength (psi)	Yield Strength .2% offset (psi)	Per cent Elongation in 1 inch
No. 1	70,000	29,000	34
No. 2	65,000	24,000	39
No. 3	78,000	26,000	28
Minimum Properties in Wrought State			
	80,000	35,000	50

The figures given in the tables show that tensile properties of materials precision cast by the lost-wax process compare favorably with wrought materials of the same composition. While the properties are somewhat lower, this is to be expected and is due primarily to large grain size resulting from slow rates of cooling. From an engineering acceptance viewpoint, control of grain size in these castings proves necessary in many applications, especially where considerable stress is involved. It is not uncommon to pull a cast round test bar in tension and find it behaving like a single crystal, the test section becoming oval shaped during loading. Since single crystals exhibit different properties in different directions, extremely large-grained cast test bars have considerable variation in properties depending on orientation of the grains. By proper control of the casting process, finer grain size can be obtained and more consistant physical properties will be achieved.

TABLE II

Properties of Type 410 Stainless (12% Cr) After Heat Treatment

Test Piece	Tensile Strength (psi)	Proportional Limit (psi)	Per cent Elongation in 1 inch
No. 1	105,000	65,000	18
No. 2	104,000	68,000	17
No. 3	104,000	65,000	16
Minimum Properties in Wrought State			
	90,000	60,000	20

Within the General Electric Co., precision casting has been devoted almost entirely to the manufacture of parts for the turbo-supercharger. Such an application requires parts that will withstand considerable stress for long periods of time at elevated temperatures. Stress-time-temperature tests to rupture are the basis for selection of materials for use in the turbo-supercharger. In such rupture testing of cast alloys, differences in grain size have shown no significant effect although short-time tensile properties may vary considerably.

Size limitations of precision-cast parts are governed by the size of the melting equipment and the quality of the investment. Most of the parts now being produced are below one pound in weight. However, pieces weighing over fifteen pounds have, on occasion, been produced satisfactorily.

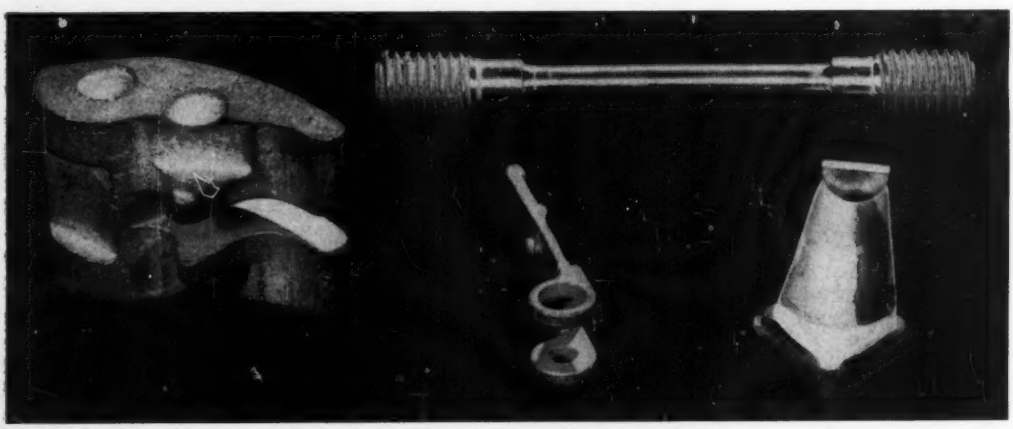


Fig. 5—Typical precision-cast parts. Two ordnance items at left are of plain carbon steel. At lower right is supercharger bucket. Part at top is a test bar

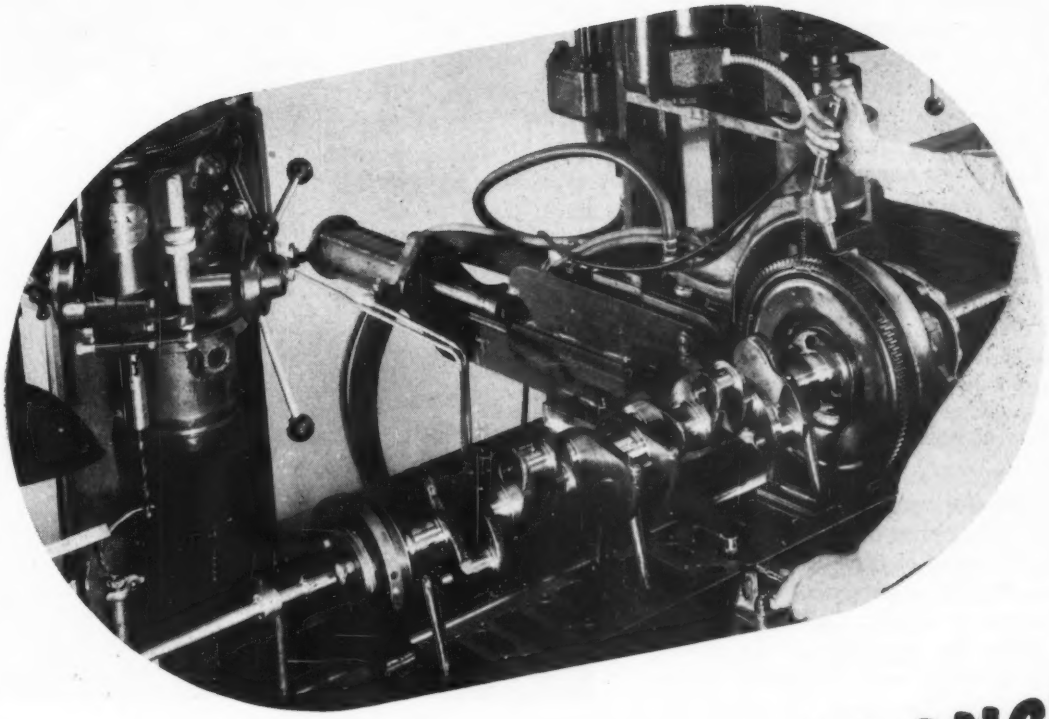
Potentialities of precision casting for future applications may be listed as follows:

1. Manufacture of parts from alloys which cannot be mechanically worked either hot or cold. Typical examples are Alnico, a powerful permanent magnet material, and many of the alloys which are used for high-temperature service.

2. Manufacture of parts from alloys which are difficult to forge, fabricate or machine. Among this class may be such items as tool bits and innumerable small stainless steel parts.

3. Manufacture of intricate shapes which cannot be made by any other method. Design engineers now have the opportunity to employ contours which heretofore it has been impossible to obtain.

4. Manufacture of limited quantities of materials ordinarily made by screw machine or die casting but where the number to be made does not justify a screw-machine setup or the cost of a die-casting die.



SPECIFYING DYNAMIC BALANCE

Fig. 17—Above—Fluid-drive automotive assembly balanced by drilling and spot welding material to the work

Part III—Methods of Correction

By W. I. Senger

Balancing Machine Division
Gisholt Machine Co.

BEFORE a rational balancing tolerance can be established, consideration must be given to the means employed for adding or removing metal for the balancing correction. Generally, the method of correction is given little, if any, consideration by the engineering department. In many cases the position of the planes of correction and the maximum amount of correction which can be applied, consistent with the strength of the body or clearance between the body and its enclosure, are in no way specified by the engineering department but are left to trial-and-error correction methods which are adopted by the production department. Such

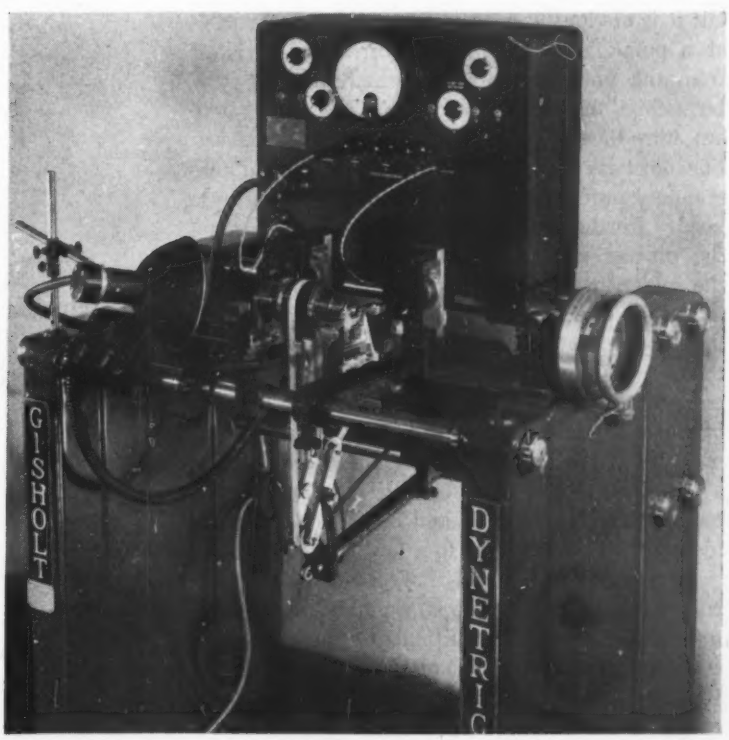


Fig. 18—Armature corrected for balance by the addition of solder

procedure is fundamentally wrong as it is the function of the drawings supplied by the engineering department to specify all limitations and tolerances so that the part may be produced easily to desired shape, strength and accuracy.

Corrections for balance, *Fig.* 17, may be made either by the addition or removal of metal. The methods commonly employed for adding weight in balancing operations are:

1. Addition of measured lengths of a given diameter of wire solder.
2. Addition of a measured length of a given cross section of strip material.
3. Addition of selected sizes or a given number of one size of known weight, made up in the form of washers, lead slugs, or cast weights.

Measured lengths of wire solder for correcting unbalance effects are commonly used on squirrel-cage fans and the rotors of direct-current motors and generators, *Fig.* 18. Corrections on the fans normally are made by adding the solder to the shroud rings, while the correction on the direct-current rotors is generally made by adding solder to the binding bands by means of which the windings on the armature are held in place. The accuracy to which a desired correction length of solder wire can be cut from a spool of this wire to desired length will determine the accuracy to which the correction for balance can be applied in production. A simple device for measuring solder lengths is shown in *Fig.* 19.

Tolerance Must Be Allowed

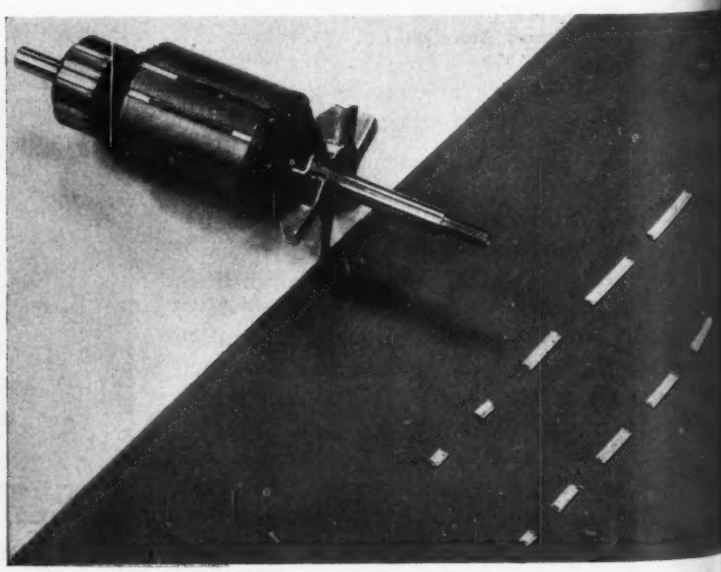
When solder is applied to an unbalanced part, the balancing machine will have indicated a point at which this correction material should be added, but it is obviously impossible to add the material at a point. As the applied correction deviates from the point of application indicated by the balancing machine, inaccuracies of correction are introduced for which reasonable compensation must be made in specifying a balancing tolerance if production is to be maintained at a high level. Further, there will be a loss of solder in the process of application due to splattering of the solder, or by the soldering iron actually holding on to some of the correction material during the operation of applying the solder. From these considerations, it is evident that, if such a correction method is specified and if a high order of balancing is required, it will be necessary to measure the unbalance in the work piece, apply a correction, then measure the remaining unbalance and apply a second correction in order to reduce the unbalance to a low limit.

Experience has shown that, for rather modest correction weights (that is, weights of approximately ½-ounce), it is difficult to apply the correction within 5 per cent of the desired amount. If the amount of correction required is in the neighborhood of two ounces, it is difficult to apply the correction within 10 per cent of desired amounts; while if the amount of correction is in the neighborhood of 1/10-ounce, the correction accuracy will undoubtedly be in the neighborhood of 2 per cent of desired value. If, in the transverse planes in which it is possible to make corrections they cannot be made throughout 360 degrees, additional balancing allowances should be provided to make possible a maximum of production. If maximum accuracy is required, then it must be assumed that three and perhaps four applications of corrections should be made to reduce the unbalance to a minimum.

Fig. 19—Device which will measure wire solder lengths required for balancing a given production part

Strip material is used for correction purposes on small universal motors as shown in *Fig.* 20, on squirrel-cage fans, and on other parts built up of or formed out of sheet material. In correcting universal-motor armatures, either extruded or strip brass, or alum-

Fig. 20—Armature for vacuum sweeper motor is corrected for balance by adding extruded strips in the core slots



inum, is inserted in the slots of the armature core to give balance. The accuracy to which the length of this material can be cut in production is approximately 1/64-inch.

Specifications Establish Tolerances

Therefore, the weight equivalent of 1/64-inch of correction material will represent the maximum accuracy to which balancing can be performed. Further, in order for the material to stay in the armature slots, a minimum of ⅛-inch of length of aluminum strip is representative of the minimum length of the lightest weight of material which can be added. Such minimum specification will necessarily establish a tolerance to which parts may be balanced unless some other correction methods are permitted.

These armatures have only a limited number of slots in the core, and if correction must be applied at a point between two slots, some error will be introduced in attempting to split up the correction between the two slots. Although it is definitely possible to chart the amount of correction to apply in slots adjacent to a given correction point for any amount of correction required, experience has shown that such charts are cumbersome and require more time on the part of the operator concerned with measuring and correcting unbalance than will be required to divide the corrections arbitrarily and make another measurement and correction for balance to come within the tolerance.

Addition of selected sizes or of a given number of one size of weight, made up in the form of washers, lead slugs, or cast weights is another form of balancing correction used where metal must be added. Weights of the cast type can only be used if the balancing tolerance is to be fairly large. The differences in actual weight of the cast parts is often as much as 10 per cent of the desired weight. This weight variation in itself will establish the accuracy to which a part may be balanced with the application of a single correction. Further error in correction is frequently introduced by the necessity for drilling a hole in the part to be balanced for a rivet to be used in attaching the correction weight. Varying rivet lengths are used for attaching varying thicknesses of weight and the size of hole drilled and the amount of countersinking used for the rivet head also is a variable where production requirements overshadow requirements for accuracy. These variables introduce errors in correction accuracy and therefore must be considered when a balancing tolerance is established. Holes drilled and tapped in the work piece to be used in attaching correction weights will introduce errors of a similar sort. Allowances must therefore be made in the specification.

Weights in the form of washers punched out of sheet steel are more desirable for weight addition. Weight variation from standard is small if dies are properly maintained. Variations in amount of applied correction will still vary with the methods used for attachment and the balancing tolerance will be established by the variation and the smallest washer weight or weight differential provided for the man who is performing the balancing operation.

Method Facilitates Correction

Frequently on rotors for alternating-current motors, rivets projecting outwardly from the cast ends and positioned between the fan blades are provided for a means of adding correction washers. This arrangement is shown in *Fig.* 21. Although providing only a limited number of correction points, it permits the measurement of unbalance, correction and checking for balance with only one handling of the work piece in the balancing machine. It gives a means for obtaining a good balancing accuracy easily and quickly.

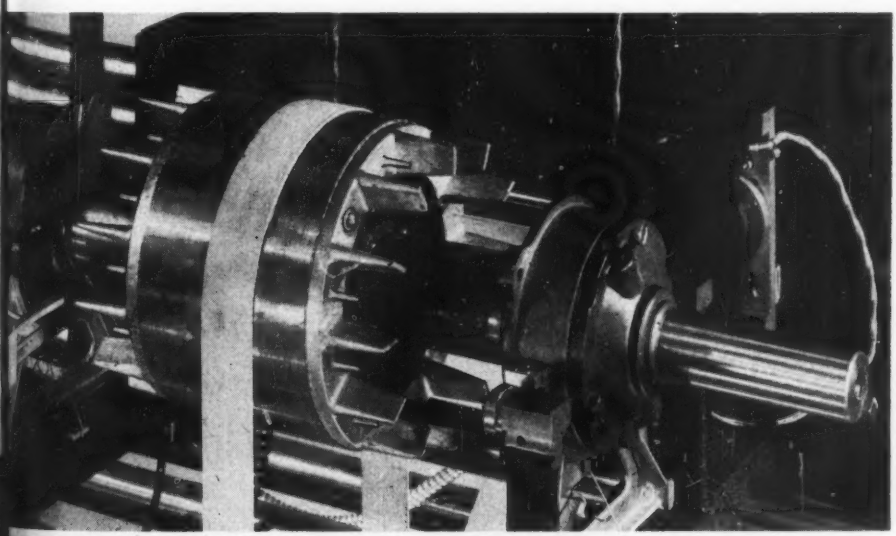


Fig. 21—Induction motor armature corrected for balance by the addition of washers

Methods generally employed for applying balancing corrections by the removal of weight are:

1. Removal of metal by milling or shaping.
2. Removal of metal by drilling.

Removal of metal by milling or shaping is expensive and requires the use of machine tools. If exact amounts of metal are to be removed, the fixtures for holding the work piece are usually elaborate. These correction methods do not lend themselves well to use where the material is to be removed from forged or cast surfaces as the depth of cut on successive parts will require measuring from points determined by the variations in the forging or casting. If the operator does not make each zero setting carefully, the accuracy of correction will be impaired. As this correction method is used generally where the depth of cut is small with respect to the area of the surface from which metal is removed, small errors in the setting of the zero position have great influence on the ultimate accuracy of correction.

Probably the cheapest and most effective method of applying a correction is by drilling. The machine tool re-

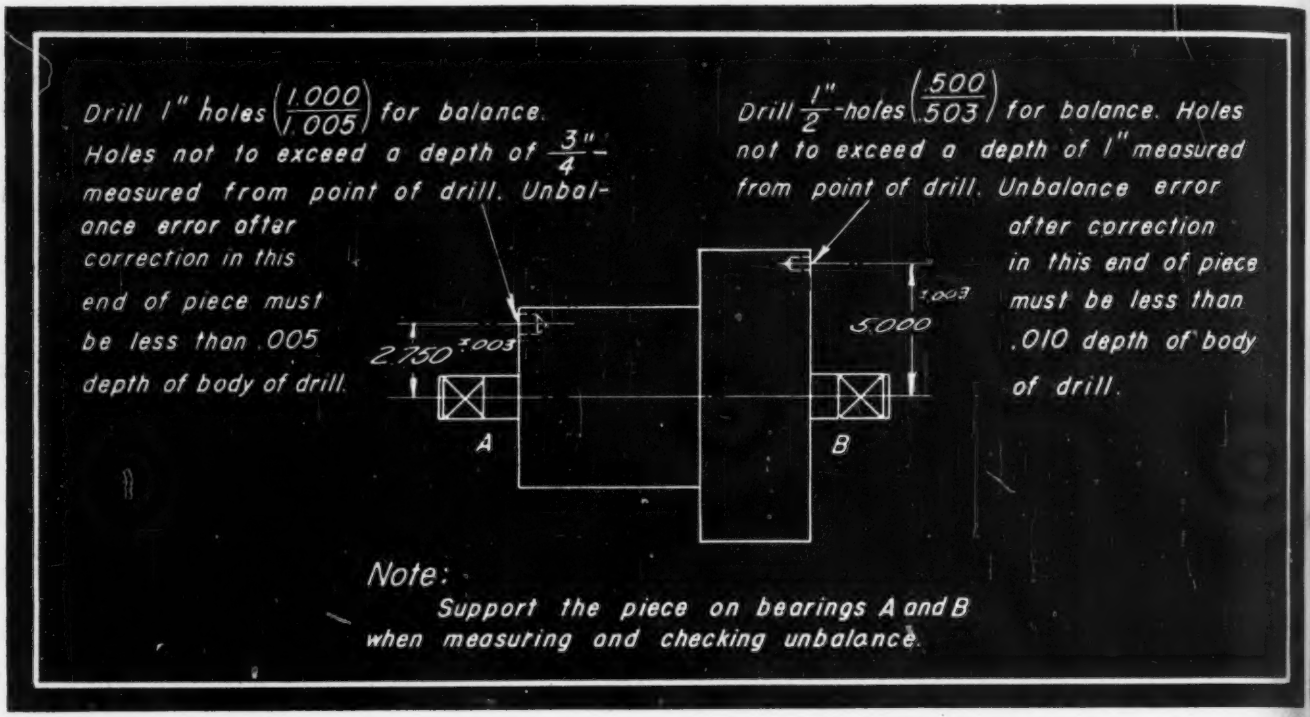
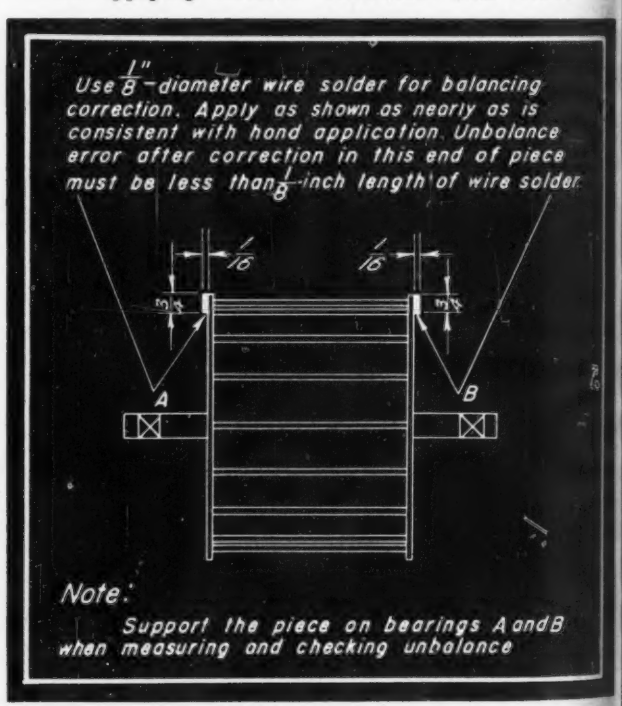

Drill 1" holes (1.000 / 1.005) for balance. Holes not to exceed a depth of 3"/4 - measured from point of drill. Unbalance error after correction in this end of piece must be less than .005 depth of body of drill.
Drill 1"/2 - holes (.500 / .503) for balance. Holes not to exceed a depth of 1" measured from point of drill. Unbalance error after correction in this end of piece must be less than .010 depth of body of drill.
2.750 +.003
1.003
5.000
A
B
Note:
Support the piece on bearings A and B when measuring and checking unbalance.

Fig. 22—Above—Detail instructions on production drawing for producing desired balance by drilling

Fig. 23—Below—Instructions on production drawing for applying balance corrections with solder


Use 1"/8 - diameter wire solder for balancing correction. Apply as shown as nearly as is consistent with hand application. Unbalance error after correction in this end of piece must be less than 1/8 - inch length of wire solder.
1/16
1/16
A
B
Note:
Support the piece on bearings A and B when measuring and checking unbalance

quired is comparatively inexpensive and the work-holding fixture and guide bushings are equally inexpensive. The zero or starting point for a drilling operation is easily and quickly set by the operator even though this point may vary from one work piece to the next. There are many drilling machines which will drill a hole to a set depth with good accuracies. The writer has seen such a machine used for balancing correction which will drill one-inch holes in steel to set depth within .003-inch, even though the point at which depth measurement starts varied as much as ½-inch. In other words, this correction device would remove a given weight of material required to give balance within an accuracy of .3 per cent on a one-inch depth of hole, regardless of the point at which the drilling actually started. Corrections of this accuracy are comparable with the accuracies of measurement which may now be obtained with balancing machines and are therefore greatly to be desired if accuracy of balance is to be obtained cheaply.

Care in Drilling Is Necessary

However, when corrections for balance are to be made by drilling, the drill should be frequently and correctly sharpened so that it will not cut oversize. The drill also should be guided to remove the danger of drilling an oversized hole. Frequent checking of the size of hole by means of a plug gage will indicate the time when the drill should be reground if correction accuracy is to be maintained. A one-inch drill poorly ground and improperly guided can readily make a hole 1.015 inches in diameter, and this variation will give a three per cent error in the amount of metal removed by the body of the drill.

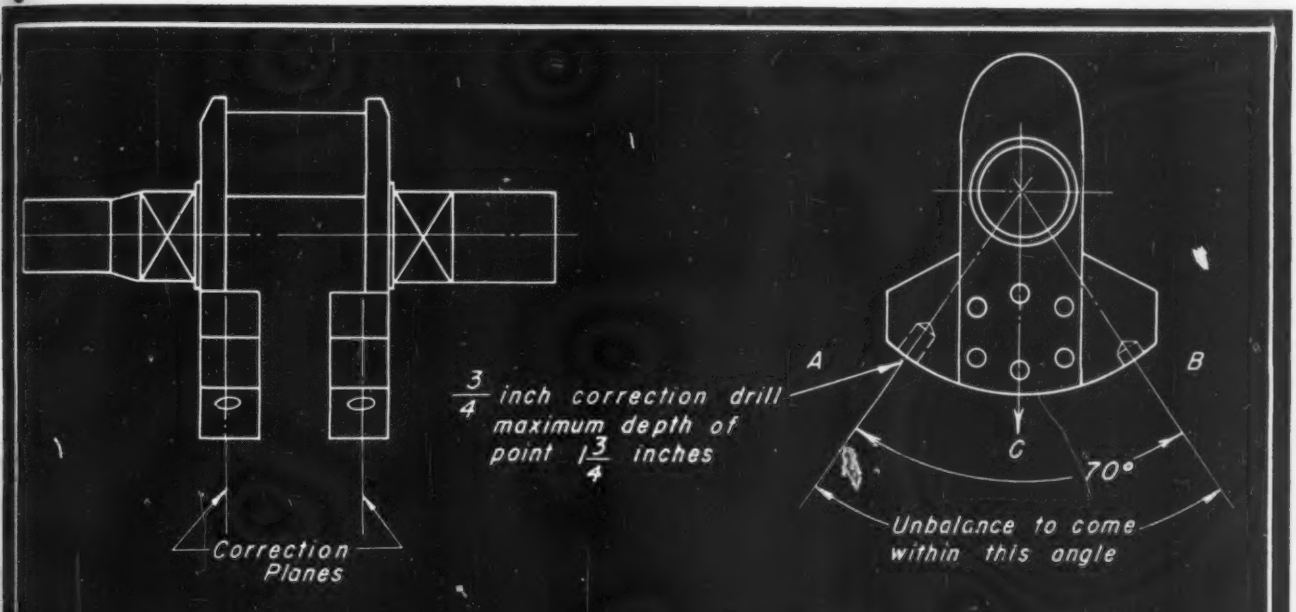
From Equation 11 (M. D. Nov., Page 106) an understandable balance and tolerance may be specified, and from the several considerations just given, it is possible to select a correction method which will permit an accuracy consistent with the tolerance. This tolerance should be established as being consistent with the physical dimensions and properties of the body to be balanced.

Drawings in *Figs.* 22 and 23 are suggested forms to be followed in specifying the method and accuracy to which balance is to be held. *Fig.* 22 gives information for a correction process by metal removal while *Fig.* 23 shows the information required where correction material is to be added. It should be noted that in each case the bal-

ancing information is given to the man performing the balancing operation in a manner understandable to anyone accustomed to normal machining operations or shop practice. It should be noted also that the tolerance is given in this same generally understood language. The information given in each case consists of the following:

1. Location of the transverse planes in which corrections are to be made. These are indicated by actually showing on the drawing the applied correction.
2. Radius at which the correction is to be applied, with a tolerance on this dimension if such is practical.
3. Method of correction to be used; that is, for example, drilling or adding solder.
4. Size of the correction device; for example, the diameter of drill or the size of wire solder to be used.
5. Limits of the correction device; in the case of *Figs.* 22 and 23, the maximum depth of drill or the height the solder may project above the surface of the work piece.
6. Tolerance to which balance is to be held in terms of the correction method employed.
7. Surfaces from which the balancing operation is to be performed. These surfaces should be the ones

Fig. 24—Radial-engine crankshafts are corrected in accordance with the indications taken from a component chart which interprets the readings of the balancing machine, greatly facilitating production operations



Correction Chart For Radial Crankshaft

Balancing machine chart for converting single measurements of the amount meter into component corrections 70° apart. The numbers are the depths in 1/64-of an inch that a 3/4-inch hole should be drilled measured from the point of the drill. Point of drill ground to a 118° included angle.

Angle Reading 0° to 35°		Unbalance In Ounce Inches																							Angle Reading 35° to 70°	
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23		
0°	A	11	19	25	31	36	41	47	53	59	64	70	76	83	89	96	103	109							B	70°
	B	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0							A	
4°	A	11	18	25	31	36	40	46	51	57	63	68	74	81	87	93	100	106	112						B	66°
	B	1	2	3	4	5	6	6	6	7	8	9	10	11	11	12	12	13	14						A	
8°	A	11	18	24	28	34	39	45	50	55	61	65	72	78	84	90	96	102	109						B	62°
	B	1	3	5	6	8	10	11	12	14	15	17	17	18	19	20	21	22	23						A	
12°	A	10	17	24	28	33	38	43	48	54	59	64	69	75	80	86	92	98	104	110					B	58°
	B	2	5	7	10	12	14	15	17	18	20	21	23	24	26	26	28	29	31	32					A	
16°	A	10	16	22	28	32	37	42	46	52	56	61	66	71	77	83	88	93	99	105					B	54°
	B	3	7	10	12	15	17	19	21	23	25	26	28	30	31	33	34	36	36	39					A	
20°	A	9	16	22	26	31	36	40	44	49	54	59	63	67	73	78	83	89	94	99	105	110			B	50°
	B	5	8	11	15	17	20	22	25	27	29	31	33	34	36	39	40	43	44	47	49	51			A	
24°	A	8	15	20	25	30	34	38	42	47	51	55	62	64	69	73	78	83	88	93	98	103	108		B	46°
	B	6	10	13	17	20	22	25	28	30	33	37	39	40	42	44	47	49	51	54	57	59	62		A	
28°	A	8	14	20	24	28	32	36	40	44	48	52	56	60	64	68	73	77	82	86	91	96	100	106	B	42°
	B	6	11	15	19	22	25	28	31	33	36	39	42	44	47	50	53	56	59	61	64	67	70	73	A	
32°	A	7	14	19	23	24	30	33	37	41	45	48	52	56	60	63	67	71	75	80	84	88	92	96	B	38°
	B	6	12	16	21	23	27	30	34	37	40	43	46	49	52	55	59	62	65	69	72	75	79	82	A	
35°	A	5	13	16	22	25	29	32	35	38	43	46	49	52	56	59	64	67	71	74	78	82	86	89	B	35°
	B	6	13	16	22	25	29	32	35	38	43	46	49	52	56	59	64	67	71	74	78	82	86	89	A	

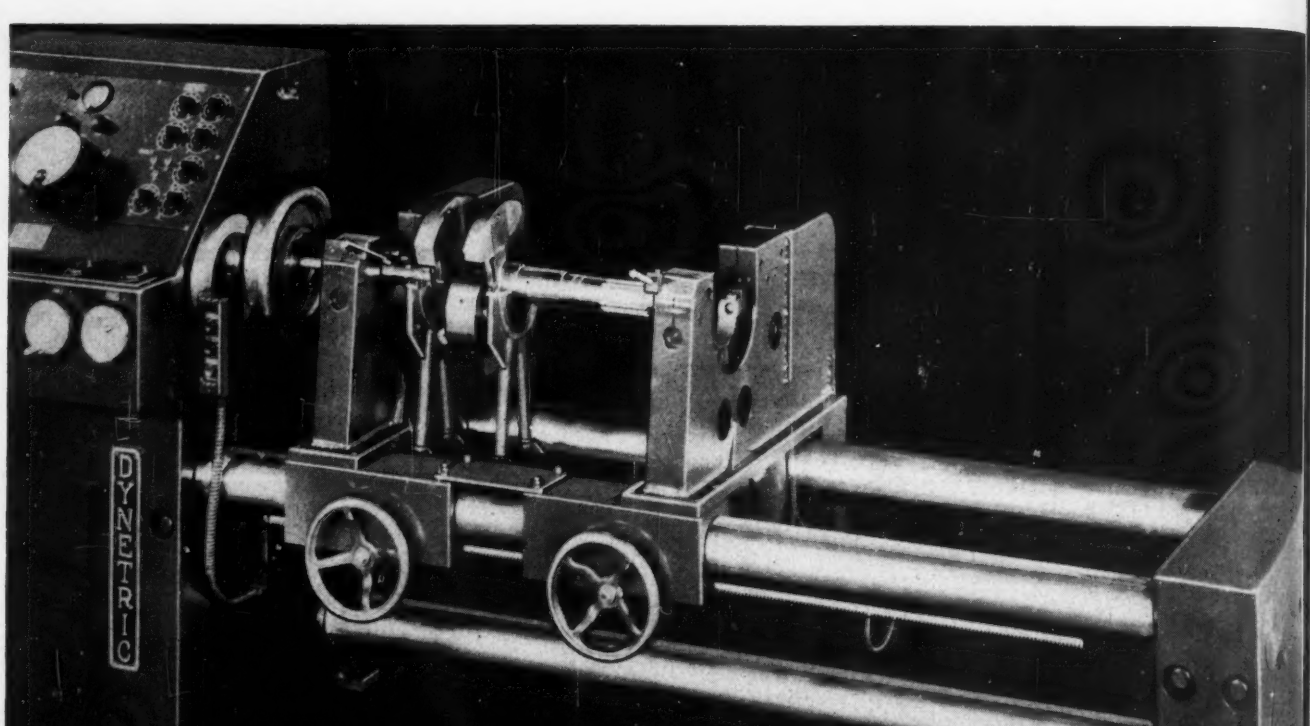
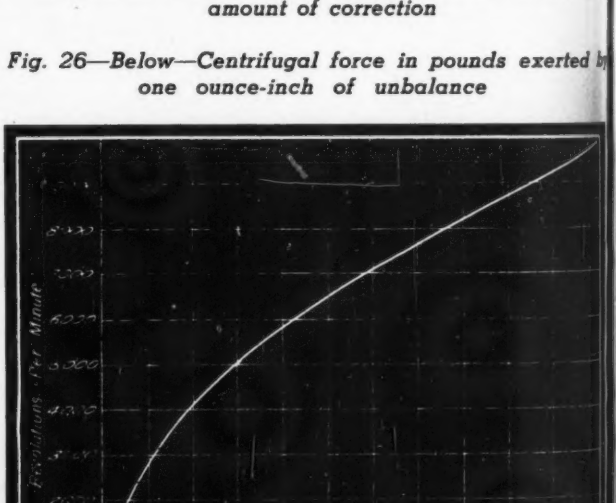


Fig. 25—Above—Because corrections on radial-engine crankshaft must be applied within an arc of 70 degrees a component chart is included on drawing to interpret amount of correction

Fig. 26—Below—Centrifugal force in pounds exerted by one ounce-inch of unbalance



which will ultimately support the part in its final assembly in the machine of which it is a part.

With a specification and information given in this manner the engineer or designer can be assured of obtaining the results desired in the elimination of vibration provided the manufacturing division has equipment capable of giving the desired results. Before entering into a discussion of the equipment available for measuring unbalance, there are a few special balancing problems which occasionally present themselves. These problems warrant mention and therefore will be treated in the following discussion.

Considerations When Balance Points Are Limited

Frequently the shape of a part to be balanced is such that correction cannot be applied at any point throughout 360 degrees in a selected plane of correction. Automotive crankshafts, fans and radial-engine crankshafts present such a problem. *Fig.* 24 shows a view of a radial-engine crankshaft and counterweight. In such parts corrections for balance may generally be made only at points *A* and *B* because the rivets used to attach the counterweight to the shaft make drilling at other points in the counterweight impossible. Obviously, corrections can be made at no point other than the counterweight. In such a case, it is general practice to make the counterweight deliberately heavy so as to introduce unbalance in the general direction of *C*. Then the amount and angular location of the unbalance are determined on a balancing machine, angle being measured from a radial line through correction *A*. By means of a component chart (*Fig.* 24), the actual corrections to make at *A* and *B* to compensate for the unbalance are indicated. Special balancing machines have been developed and are in use which will measure directly the correction required at *A* and *B*. Such machines, *Fig.* 25, will be discussed in Part IV in the next issue.

The question, "Shall I specify static (one plane) balance or dynamic (two plane) balance?" is often propounded. This question may only be answered by careful consideration of a number of elements. Generally, if the part to be balanced is essentially a disk (its thickness small in comparison with its diameter) static balance will be satisfactory. For all other parts dynamic

balance is required. If the part under consideration runs at low speed (a few hundred revolutions per minute) static balance will generally be satisfactory even though the thickness of the part approaches the diameter of the part.

There are some further general considerations which must be given attention in determining requirements for balance. The first of these is the rotational speed of the work. From the curve of *Fig.* 26, it is obvious that the centrifugal force due to a given unbalance increases rapidly as the speed increases—as a matter of fact, it increases with the square of the speed. Therefore, a supercharger impeller rotating at 30,000 revolutions per minute must be very accurately balanced even though the corrections must be applied by a rather difficult method such as by use of a small hand grinder or by utilizing a fine-cut file.

Mass of Machine Determines Degree of Balance

Another general consideration in determining upon an accuracy of balance is the mass of the machine into which the rotating body is mounted. If bearing loading and bearing wear may be neglected, a rotating body mounted in a machine having a total weight ten times as great may have a balancing tolerance five times as great as that of a rotor mounted in a machine having total weight twice the weight of the rotor (if rotational speed and all other factors are equal). This is obvious from the fact that the ratio of weights of the two machine types is in a ratio of

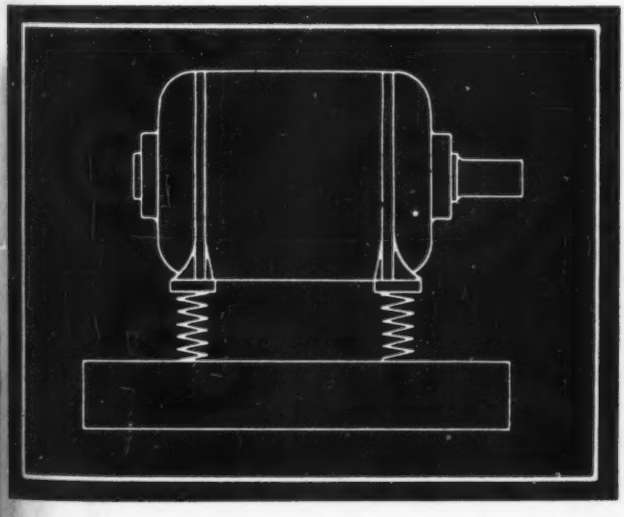


Fig. 27—Natural frequency of mounting should be selected so that it will not be in resonance with rotating parts

five to one and therefore the vibration of the complete machine due to a given unbalance will be one-fifth as great in the heavy machine as in the light machine.

Frequently, well balanced parts are mounted in a structure which will vibrate considerably when the well balanced part is rotated. This vibration can be caused by a portion of the structure supporting the rotating elements having a natural frequency the same as the rotational frequency of the rotor. Such vibrations are easily eliminated by changing the natural frequency of the supporting structure. A simple demonstration of this condition may be made in the following manner: A variable-speed motor is mounted on springs such as illustrated in *Fig.* 27.

With the motor running slow, speed is increased gradually by small increments. At some speed a considerable vibration of the motor will be observed even though the rotor is reasonably well balanced. If the supporting springs are changed to either a weaker or stronger set, the motor mounted on the new springs will not vibrate if the rotational speed is maintained the same as it was when the vibration was observed on the original springs.

Design Should Incorporate Balancing Provisions

In this discussion, considerable mention has been made of balancing tolerance and practical methods of applying correction. Consideration must also be given to the maximum correction for which provision must be made. Large corrections should be provided for to insure that every part may be balanced by the method prescribed. Ap-

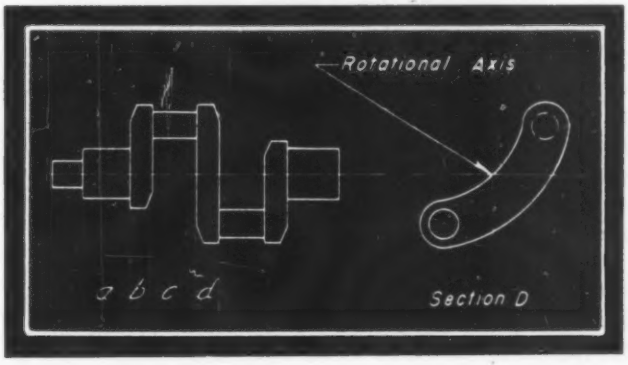


Fig. 28—Division of crankshaft into elements to determine requirements for balance correction

proximation of the amount of correction required may be made in the following manner. *Fig.* 28 represents a crankshaft for which it is desired to determine the maximum requirement for correction. This body may be divided into a number of transverse sections, *a*, *b*, *c*, *d*, etc., the sections being such as have the same characteristics throughout their length (measured along the rotational axis).

Sections like those at *a* and *c* which are machined all over can introduce only small unbalance effects and need not be considered in this rough approximation. However, a section such as *d* which is not machined, can introduce considerable unbalance. It is reasonable to assume that the center of gravity of this section may shift from crankshaft to crankshaft as much as .01-inch due to wash of the forging dies, trimming of the forging, distortion in cooling after annealing and in centering the crankshaft forging. Therefore, the unbalance variation which can be introduced by this section is .01 times the weight of the section in ounces. If all variations of such sections are added together, an approximate figure representing the balancing correction is obtained. When this summation has been obtained, provision should be made for the addition or removal of at least 50 per cent more than this estimated amount of correction material.

(*Concluded in next issue*)

How an Industry Looks Ahead

INDICATING the type of planning necessary for reconversion including the time required for engineering, testing, tooling and purchasing of materials before actual production, the automotive industry's chart, below, prepared by the Automotive Council for War Production shows the basic time elements which must be taken into account.

While plants are putting war work first as should be done, they are making plans to bridge the gap between war contract terminations and production. Blueprints must be ready to meet unexpected shifts in the production picture, even though the automotive industry as a whole is pressing its war production program forward at a ten-billion dollar a year rate, with no let-up in sight.

Planning May Obviate Suffering

How much the nation's civilian economy suffers during reconversion will depend to a considerable degree upon the length of time industry takes to re-establish its production pattern which was torn asunder and strewn widely in the wartime changeover. Lack of machinery, facilities or materials can block the flow of production and deprive men of jobs all along the route.

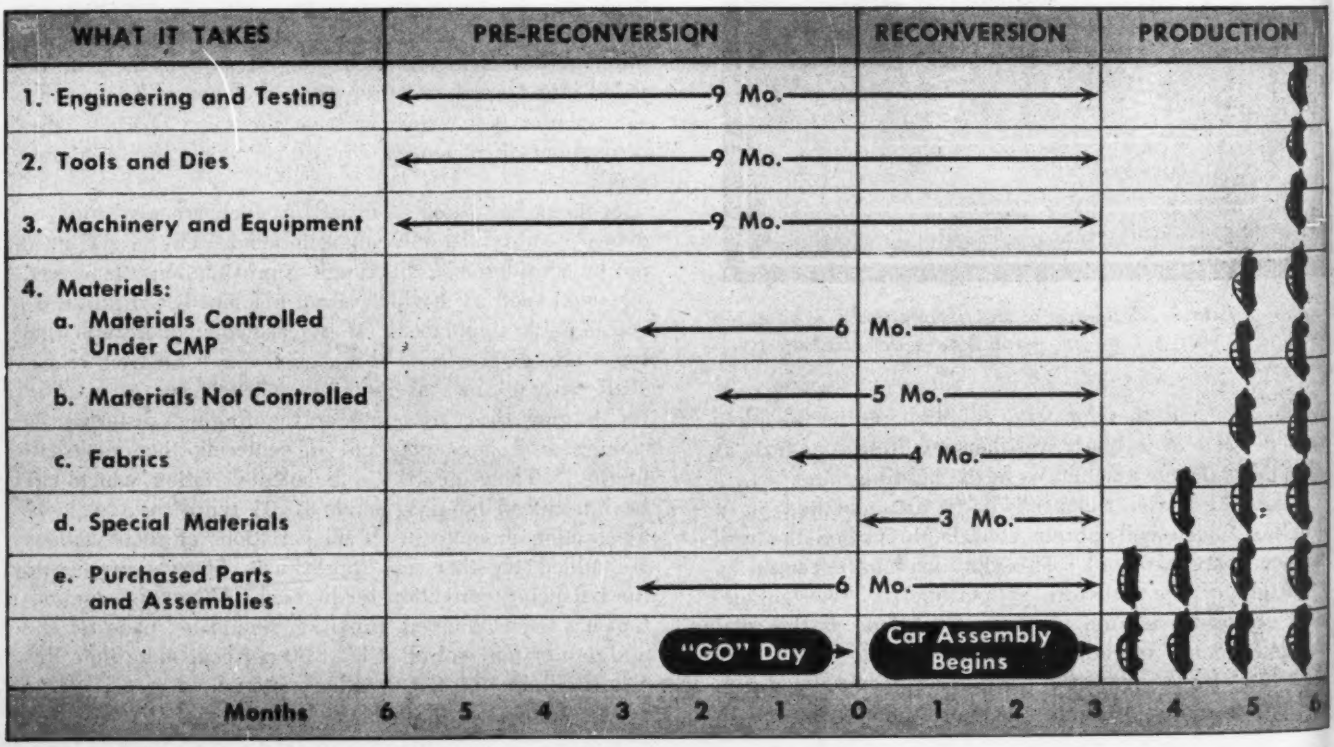
One automobile company estimates that it needs 3600 new machine tools, as replacements for tools lost at the time of conversion, to enable it to resume production at the rate of 50 per cent of its 1941 schedules. Since it takes months to design and build such machines and sometimes weeks to set them up for operation, the time involved in this phase of preparation is a fairly fixed factor. Another car manufacturer has been notified that some of his bottleneck machines cannot be delivered for many months. Similar conditions are faced by other manufacturers.

In order that part makers will know the prime manufacturers' needs in advance, production drawings and material specifications are now being brought up to date. The War Manpower commission recently authorized the transfer of a limited number of technicians for such work.

Still other factors that must be planned for are the torrents of problems that will flood in on management when war contracts are canceled wholesale. One motor company calculates that, in its plants alone, about fifty-million dollars worth of Government-owned equipment and 100,000 tons of all kinds of material in every stage of refinement from rough billet to finished assembly, will have to be cleared out of some 17-million square feet of floor space for complete reconversion.

The race to get cars out will find all motor companies striving to telescope time, with a variety of shortcuts being employed. Because no two companies face the same problem, a wide time gap may separate the fastest producer from the slowest in getting the first cars on the market.

The accompanying chart graphically indicates the planning and timing necessary for all activities leading up to automobile production. Previous to "Go" Day or the actual beginning of reconversion, six months must be allowed for engineering and testing, tool and die making, and procuring machinery and equipment. Other activities require less time, as shown. From the start of production, three months will be required to reach full schedule.



Trends in Design of Weldments

By Edward J. Charlton
Lukenweld Inc.

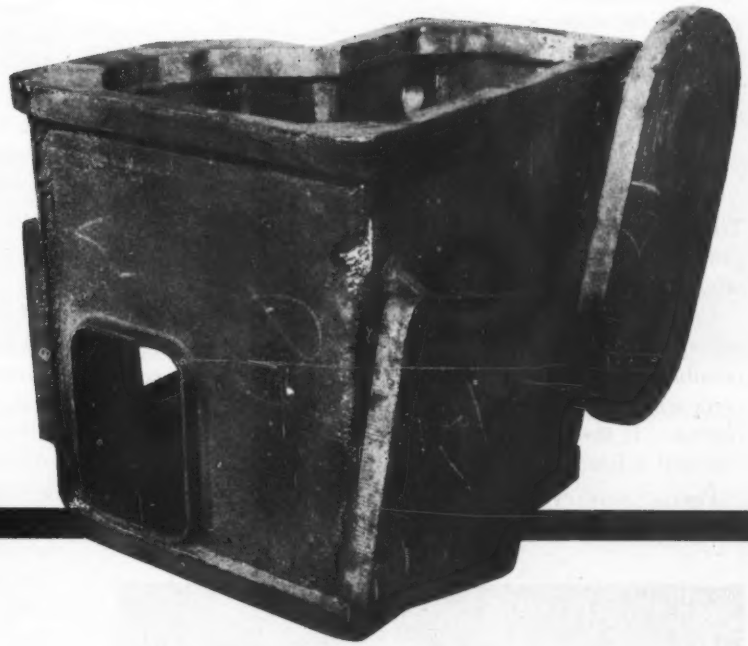


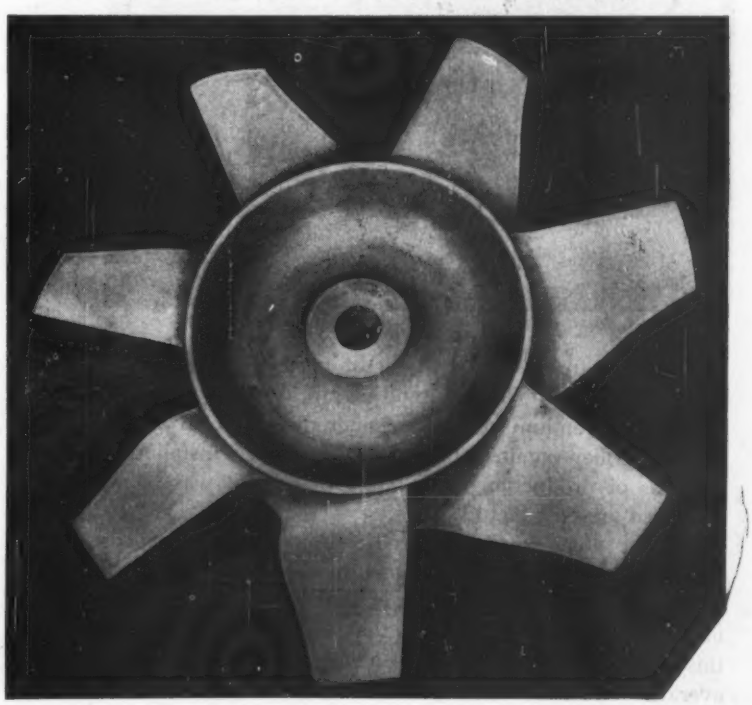
Fig. 1—Above—Although produced by the hundreds, this gear housing is made as a weldment because of high predictability

USE OF welded machinery parts is about to enter what is perhaps the most important stage in its growth. Records based on approximately 15 years experience have served to prove the adequacy and engineering value of such parts in diverse fields, and certain convictions or assumptions in design philosophy have been substantiated, particularly under operating conditions embracing fatigue and impact.

Basic impetus behind this general acceptance can be divided into two parts—economic and engineering. An additional impetus during the war years has been the abnormal urgency of acquiring necessary parts by any means as quickly as possible. In many such cases, normal economic justification must have been nonexistent. Undoubtedly much misuse of weldments has resulted because of war necessity. Instances exist of their application solely because of scarcity of other materials, while criteria established by battle damage possibly have introduced the use of weldments in applications not otherwise justified. Inevitably, in instances of this nature, the trend will be toward reversion to the use of castings.

Without doubt certain changes in design criteria are due, for weldments have been applied to machines the overall design of which was frozen for one reason or another, with the result

Fig. 2—Below—Severe operating conditions due to high speed dictated the use of welded forgings for this blower impeller



From a paper presented at the recent annual meeting of the American Society of Mechanical Engineers in New York.

that possible superior advantages have been submerged. Unless the machines are redesigned to realize the full value of welded construction, reversion might occur in these cases also.

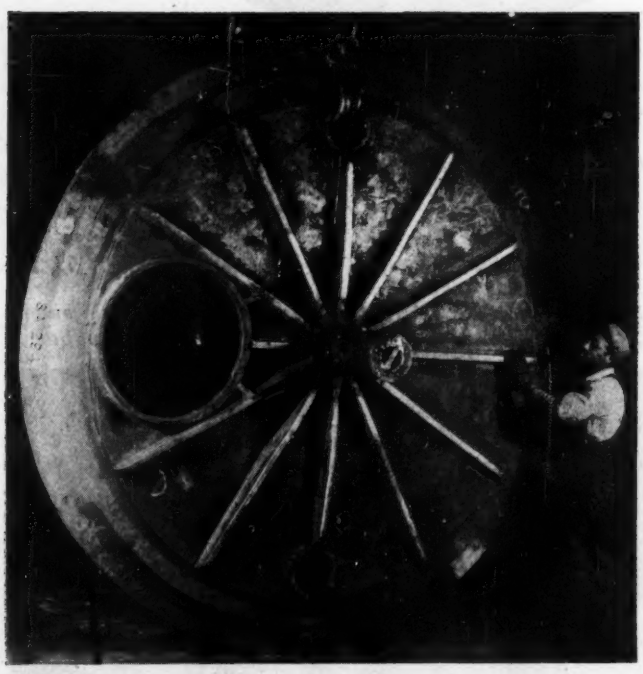
Several basic reasons for the use of welded parts apply also to the selection of any engineering material, and reflect either economic or engineering concepts or a combination of the two. These reasons are:

1. Comparative first cost
2. Predictability
3. Strength characteristics
4. Need for greater rigidity
5. Weight reduction
6. Wearability
7. Operating efficiency
8. Natural adaptability.

These factors determine the selection of a material or process, and usually one of them dominates, with the others as lesser influences.

COMPARATIVE INITIAL COST loses its engineering significance if the design is adequate, if it is economical, if possible indirect economies have been considered in cost comparisons, and if the specifications are clear and conclusive. It then becomes purely an economic consideration and a function of the purchasing department.

PREDICTABILITY can be a matter of insurance. Such insurance is an economic reason justifying the use of a new



type of structure. If the first cost of the weldment exceeds that of a previous method of fabrication which had failed, the premium can be considered as that on an insurance policy against failure. Machining, installation and loss of production while the replacement was being built also are cost factors.

Another item involving predictability is rejection or scrappage. The small part shown in *Fig.* 1 has been produced by the hundreds and it is hard to conceive that its first cost could compete with that of a casting. Use of this weldment stems from the alleged expenditure of an average of 75 hours for repair on each such casting. The user of this part apparently feels that a weldment is justified because of its higher degree of predictability, and a report on the repair time states that the alleged 75 hours previously necessary have been reduced to one or two hours on the weldment.

All these factors involving predictability in the trend of weldment use have only economic significance. A factor of engineering significance is consistent predictability required by severe operating conditions. The high-speed blower fan, *Fig.* 2, is composed of a central forging with drop-forged vanes welded to it. These fans operate at 5000 revolutions per minute, are tested at 7500 and have been used for several years prior to the war. Obviously the inherent soundness of the forged and welded steel is considered insurance against failure which would be very serious at the speeds indicated.

Incorporating Castings in Weldment

COMPARATIVE STRENGTH CHARACTERISTICS of competing materials are the third reason behind the trends under discussion. *Fig.* 3 shows the end portion of a rotary mixing unit, many of which are in operation in a chemical process. The original design used steel castings and properly so, for at that time weldments as machinery parts did not warrant consideration since too little was known about their design. The castings failed consecutively, following a limited service life. Repairs were attempted unsuccessfully so that replacement became a normal procedure. The difficulties of shape and size limitations apparently were such that a steel casting of the desired strength was not possible in this case.

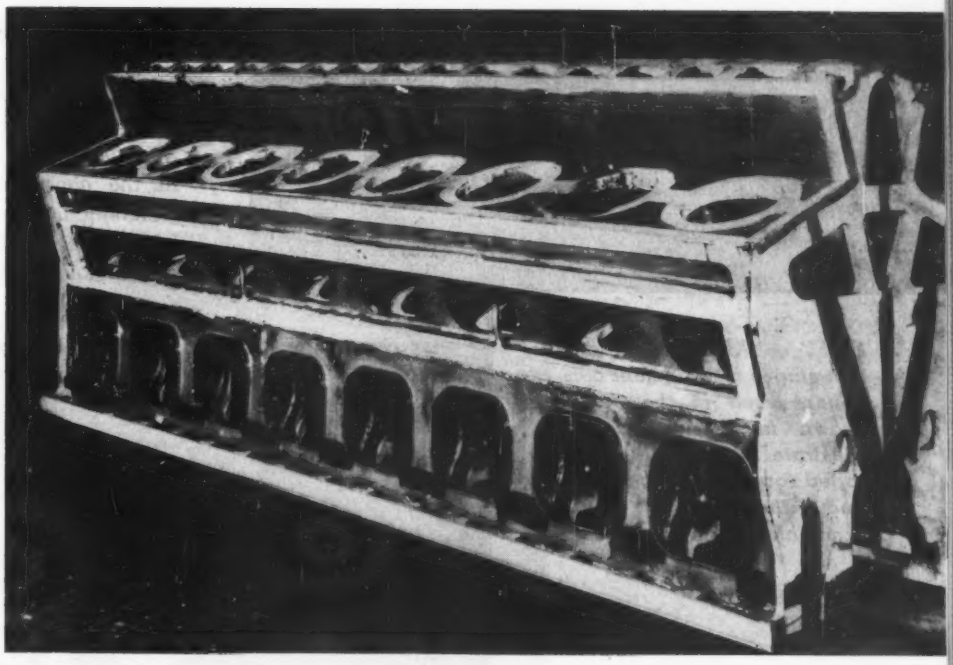
Finally the user considered a weldment. The piece shown is the result, which has now become standard in this application. The original part has been in service for over five years and no replacements have been necessary. No change in the mechanical design was needed. the weldment simply replaces the casting within the size and clearance limitations imposed.

A substantial part of the structure, however, is still made from steel castings. Castings were used in the design where economical, as well as to obtain flowing contours, which are inherent in a casting, at those points

Fig. 3—Left—End portion of rotary mixing unit has cast steel rim and hub, welded to the steel plate construction comprising the remainder of the weldment

Fig. 4—Right—Weight saving is primary consideration in use of welded steel construction for this diesel engine frame

Fig. 5 — Crankcase frame for sixteen-cylinder marine diesel is typical of numerous welded steel parts which have survived battle



where such is desirable. This redesign represents a profound difference in the concept of applying castings. The castings used in this part are of such size and shape as to be desirable and readily produced in the foundry, in contrast to the difficult production features of the entire piece as a single casting.

Two things are apparent in connection with comparative strength characteristics. First, steel plate has been substituted in the large flat expanses involved; secondly, the concept of welding has permitted the elimination of the fundamentally weak points. The piece is subdivided into small components of predictable strength, integrated with welds of predictable strength.

In such a mechanical part, a comparison of physical properties applicable to static loading conditions is not adequate. Repeated and reversed loading occurs, and the endurance limit of the structure becomes the controlling criterion. There is a trend toward making the endurance limit and impact resistance definitely specified properties of materials to be used in the structural parts of machines.

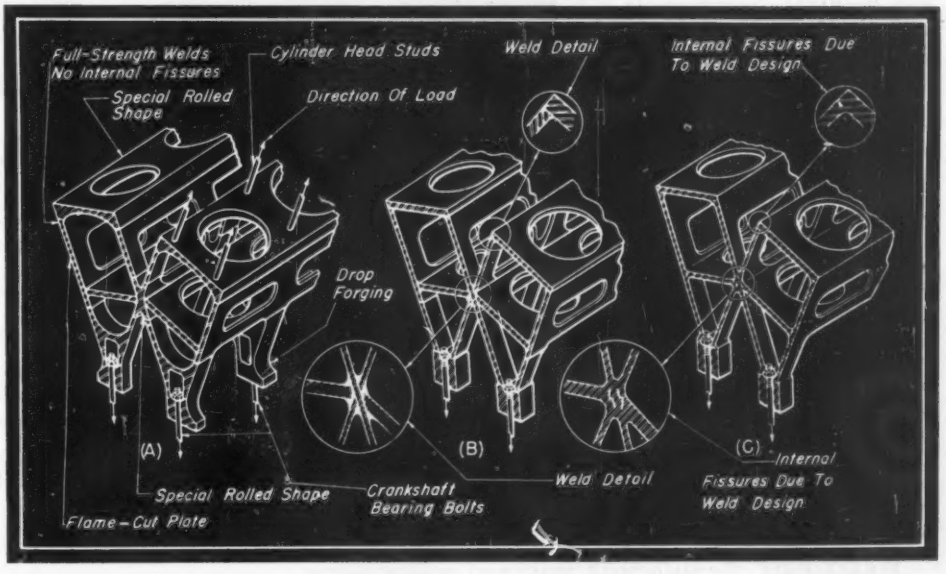
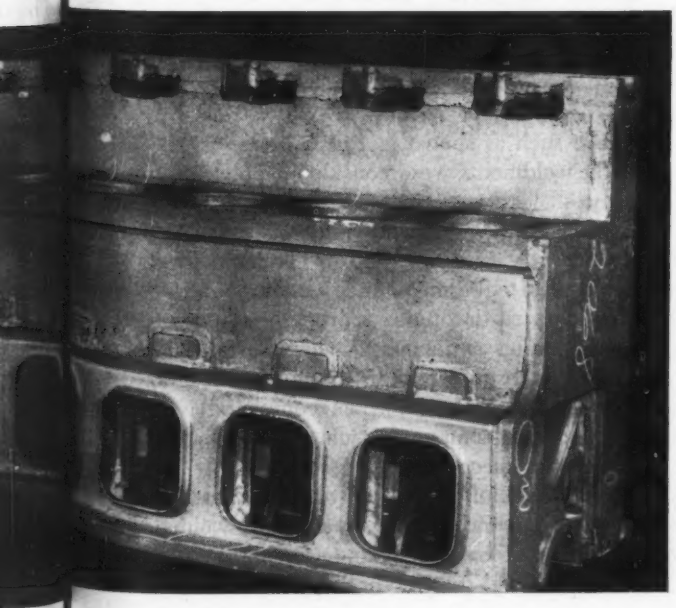


Fig. 6 — Right — Combination of forgings and special shapes in welded diesel engine frame (A) is superior to all-plate construction (B) which has angular contours and internal fissures that become evident when the design is depicted as a single piece of metal (C)



MAXIMUM RIGIDITY, to be discussed clearly, must be considered within definite weight and space limitations. Thus limited, maximum rigidity obviously can be realized only by disposing material with the highest possible modulus of elasticity in the most effective manner. Thus there is no question that an iron casting is more rigid if it weighs enough or is sufficiently larger in cross section. The foregoing remarks may seem elementary, but they are made because the repeated assertion in the early days of welded steel, that cast iron is stiffer, is still heard occasionally.

WEIGHT REDUCTION is the fifth reason in the selection of a structural part. This reason has established to a great degree the trend in the use of weldments. Weight is a

Fig. 7 — Above — Production stages of weldment component used in the design shown in Figs. 5 and 6 illustrate initial billet, rough rolled form, flame-cut shape, and the shape rough-machined to dimensions

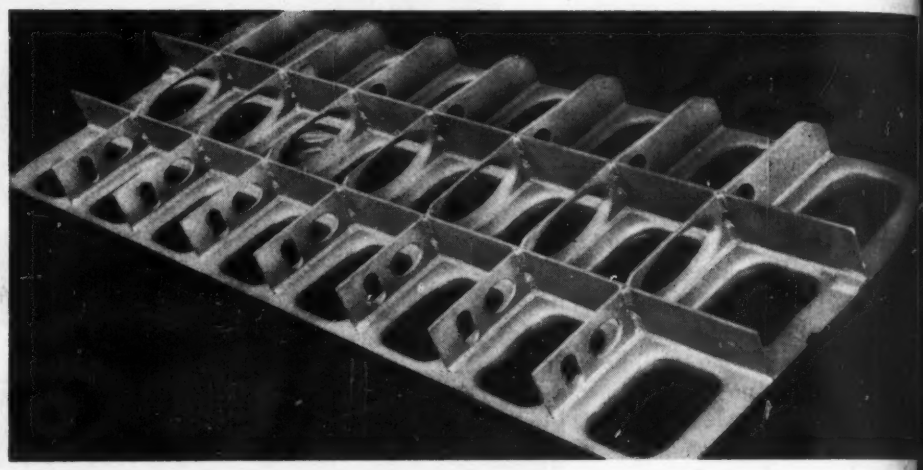


Fig. 8—Right—Subassembly, with welding completed, is accessible from standpoint of welding and inspection

watchword in transportation equipment with current emphasis on increased speed, safety and operating efficiency.

Weight reduction dictated the engineering selection of construction for the diesel engine frame shown in *Fig.* 4. This frame is more complicated than it appears. Much of it is composed of metal 3/16-inch thick, and such thin sections throughout its structure connect with heavier sections in a somewhat abrupt manner. There are labyrinths in its shape involving many small enclosed pockets. Clearly it would be difficult, if not impossible, to use any other known form of construction with less weight. Thousands of these frames have been produced without question of predictability or design hazard.

WEARABILITY, the sixth point mentioned as establishing trends in the use of weldments, is typified by the "riding rings" on rotary kilns. Formed in bending rolls to a 12-foot circle and butt welded, each ring is a plate 8 inches thick, 23 inches wide and weighing 12 tons. Wear naturally controls their service life, but they do not wear out as quickly as those previously used.

Operating Efficiency Increased

INCREASED OPERATING EFFICIENCY of the machine in which the parts are used is the seventh reason. The steam-heated dryer roll used in the paper, printing, drug, and textile industries is an important advance in the use of welded machinery parts. At least 250 welded steel dryers are now in use, some of them having been installed for more than ten years. Although they are lighter in weight than castings, this is not usually a prime consideration. In one application steam consumption was reduced approximately 40 per cent, yet production was increased. Thickness of the outer shell is approximately ½-inch on one dryer operating under a steam pressure of 100 pounds. The replaced iron dryer, with much thicker shell, carried 20 pounds of steam.

NATURAL ADAPTABILITY—the suitability of a welded structure to design requirements—is the final item establishing trends. A typical example is that of a beam used in the construction of a strongback bending roll and weighing approximately 30 tons. Roughly it is 50 feet long and 6 feet high with flanges 5 inches thick and web plates ⅞-inch thick. It seems obvious that welded construction is most suitable for parts of this nature. Clearly riveted construction would not be more adaptable or economical. It is not likely that castings in either steel or iron would be more economical.

In most applications the fundamentals of selection apply in combination. The foregoing examples are intended to illustrate the predominance of the reason under which each was described. Developments excited by wartime considerations have been avoided deliberately. The prosaic examples shown were selected, first to illustrate the varied reasons for the use of weldments, and secondly, to show the breadth and depth of their application.

Utilizing Rolled Steel

Certain engineering fundamentals underlie the use of any type of construction or material. The basic fundamental behind the use of weldments is the engineering utilization of rolled steel. Such utilization may involve simple supports or it may involve a relatively complex structure such as shown in *Fig.* 5.

Early weldments were considered practically as riveted structures. It was assumed that a welded joint was simply a joint. A pure butt weld was considered a hazard and structures were designed to eliminate them.

Electrodes, equipment and technique have developed progressively so that today a properly executed weld need not be regarded as a joint. A wide background of operating history substantiates this, for it is a recognized fundamental that a weldment when completed is not a number of separate components joined together, but a single homogeneous piece of steel.

The material, of course, must be weldable. Many types of carbon steel and its alloys are readily welded, but some require special care involving, for instance, preheating.

Field welding, or structural repair, must be taken into

account in the selection of steel. Perhaps design loads are based on physicals dependent on special heat treatment. It is practically impossible to heat treat the weldment in the field. Possibly the steel is not weldable at low temperature. Ordinarily these can be said to be considerations of little importance. They apply, however, in the selection of materials, for instance, in railroad service. In wrecks of minor nature structural parts must be readily repaired either by rewelding or straightening.

Second fundamental is that of design criteria. Naturally, the first of these is the type of service required of the structural part. The question of service requirements has been detailed in the foregoing eight basic reasons. However, a specific example, which involves many implications with regard to service requirements as a basic criterion of design is of interest.

Several years ago a submarine was commissioned, powered by diesel engines the frames of which were of welded steel similar to the one shown in *Fig.* 5. This vessel completed roughly 10,000 hours on these engines in normal service and from any standpoint the design of the engine frame had been proved satisfactory. Many hundreds of similar frames are in constant service today.

Effects of Battle Action on Weldments

This particular submarine, however, has an extremely interesting history from the engineering standpoint. We hear that she was depth charged many times, on two occasions so drastically that she was blown to the surface. Ultimately the ship arrived on the West coast with known fractures in the engine frames. Some repair was executed at that time and she was then brought to the East coast under her own power for general overhauling.

The engines were dismantled completely, permitting detailed inspection, with the result that the engine frames were replaced, many more fractures having been discovered. To broaden the knowledge of design criteria, one of these fractured frames was returned to the fabricator for careful investigation. Location, type, and possible causes of the fractures were studied and all surface cracks and every inch of weld metal were examined carefully.

Exactly 126 separate fractures were found—all of which undoubtedly occurred some time during combat or short-

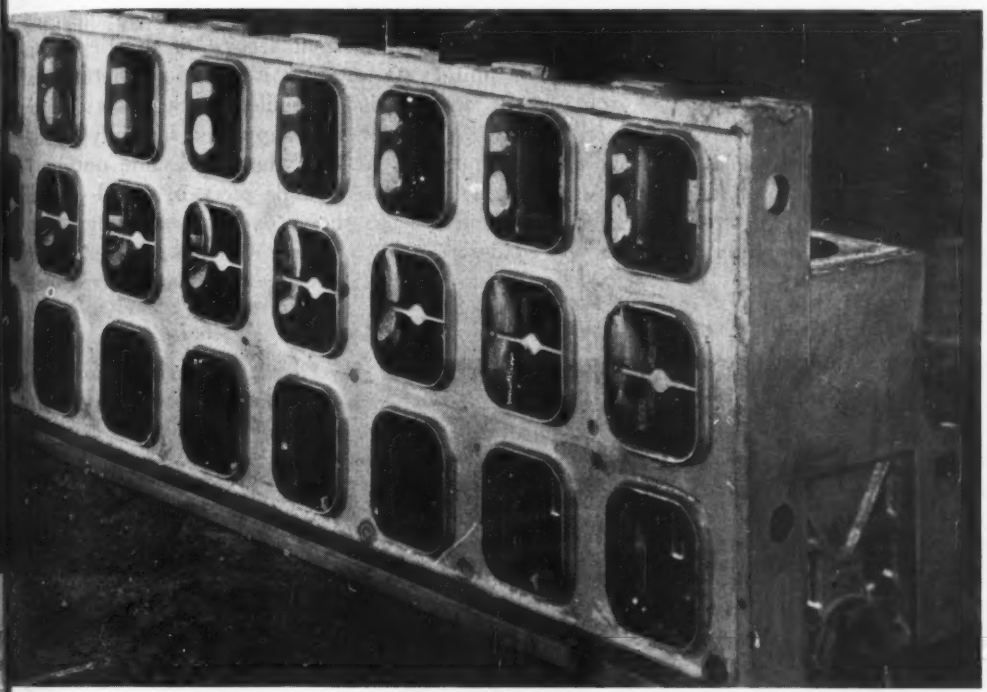
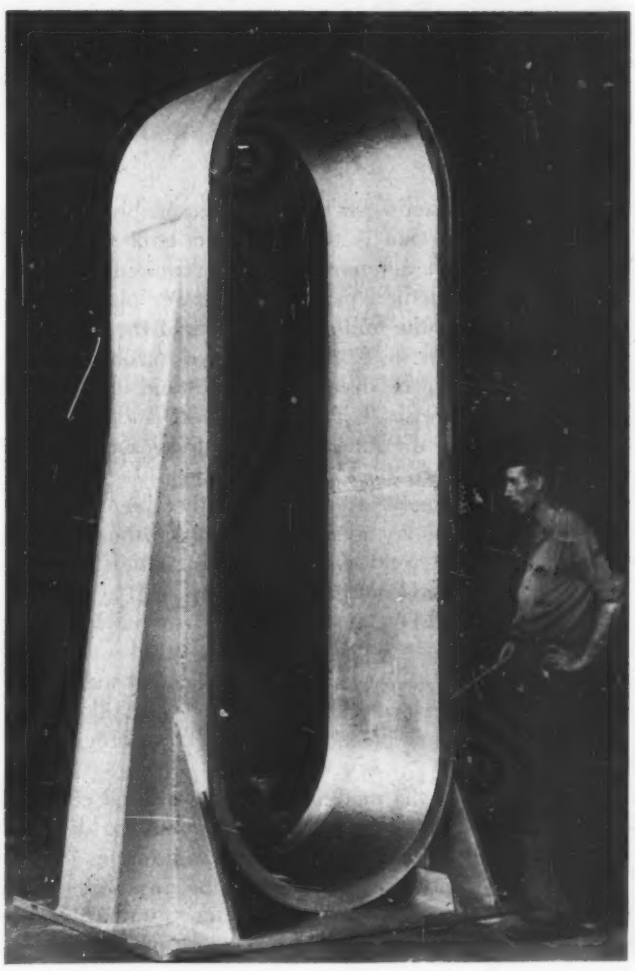


Fig. 9 — Left — Welded engine frame incorporates subassembly pictured in Fig. 8, the joining welds having been executed through the openings

Fig. 10—Below—Welded frame for 2000-ton hydraulic press consists of two steel plates formed in U-shape and joined by butt welding



ly thereafter since they were noticed while the ship was still at sea—nevertheless the ship, of necessity, traveled several thousand miles following their discovery.

With reference to type and location, the cracks can be separated into a few categories. Some of them started at stress raisers due to machined contours, others at contours about openings. The profoundly interesting finding is that no failure started in the welds, or parent metal, of primary load-carrying members. Some of these members have two or more butt welds within their length, dynamically loaded in tension.

Some failures started in welds of secondary members, welds that were deliberately designed as simple fillets with internal fissures remaining. It was assumed that the stress in them was of a type or direction that would make such joints adequate, and they *are* adequate for normal operation.

In *Fig.* 6 an outline of the primary portion of the engine frame just discussed is shown at (A). With regard to the disposition of metal, note the contours and the position of the welds. The frame could have been designed and built as shown at (B), and as a matter of fact many are so designed initially. Undoubtedly some have gone into service so designed at such points.

It is reasonable to assume that even though they survive normal service, abnormal service of drastic nature will cause failure. At any rate it is our conviction that such designs are utterly unpredictable and survive by good luck. This conviction is supported logically by the assumption illustrated in (C) where the same outline is shown as though it were a single piece of metal—which it is. The internal voids shown exist to some degree unless the welds are made properly. Welds at such points in such a design are hard to make, and their X-ray inspec-

Fig. 11 — Left — Half of a formed steel head is used to provide the vertical walls and rounded corners of the main portion of this weldment, which is part of the upper housing for a ship turbine

Fig. 12—Below—Component part of the rear axle housing shown in Fig. 13 is flame cut, hot formed and pressed

Certain conclusions can be drawn reasonably from this service history: One is that design criteria for normal operation is much different than that required for battle action, if the structural parts of the power plant are expected to survive the hull or other parts of the vessel.

Another conclusion is that a welded machinery part most definitely can be designed to withstand unusual and indeterminate stresses. Those members and joints intended to carry maximum normal stress survived the drastic shocks perfectly.

Some of the fractures starting at other points crossed over welds in primary members, which substantiates the conclusion that a welded joint can be so designed and executed that it need not be regarded as such. Otherwise the fracture would be expected to follow such a weld once it found it.

Finally, the frame was sufficiently rigid and structurally stable so that the severe shocks experienced still did not affect the operation of the engines. They brought the ship home.

If properties of the material are to be utilized efficiently, the metal throughout the structure must be disposed properly. Where minimum weight or maximum rigidity are of prime importance, metal must be disposed so that design loads are carried with the most uniform stress throughout or with the maximum moment of inertia.

Fig. 13—Below—Rear axle housing is a weldment incorporating two of the component parts in Fig. 12

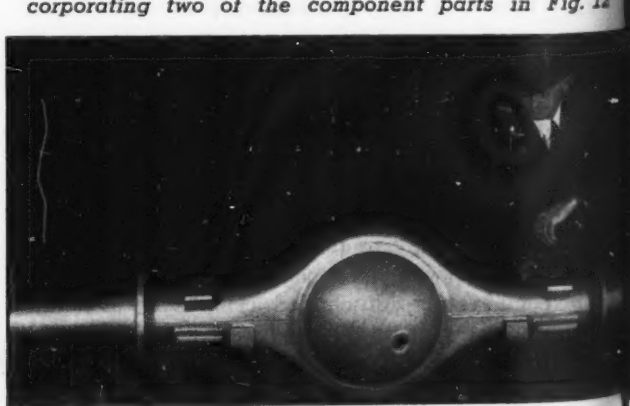


Fig. 14 — Drop-forged sections shown in foreground are flash welded to form the weldment component behind

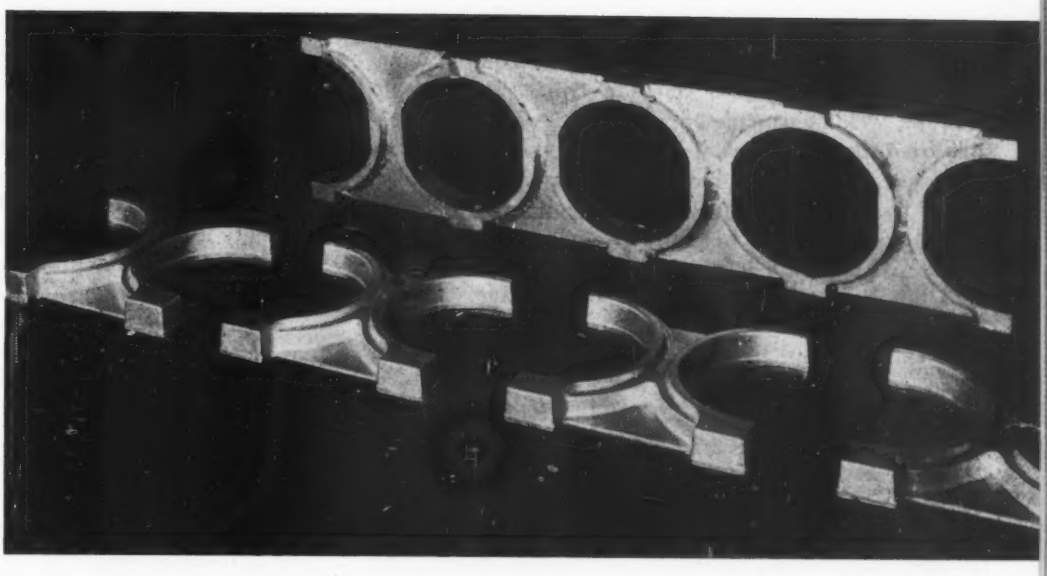


Fig. 15—Below—Eight small steel castings are flash-welded to provide this component

Fig. 16—Below—Drop-forged components are typical of many used in weldments. Machined forging at right is hub of a large gear blank

tion is difficult also.

Another point concerning proper disposition of metal is external contour. Flowing contours are natural to castings, the necessity being imposed by foundry requirements. They are necessary for another reason, too, in the structures of the type which we are discussing, because they provide relatively even stress levels. This logically is an absolute requirement for minimum weight and maximum serviceability.

Such design criteria cost money and there seems to be a natural tendency to just "weld" and hope for the best. Service records, however, substantiate our convictions on this score. *Fig.* 7 shows the production stages of the component noted in the center of the left view in *Fig.* 6(*A*). On the left is the initial billet. Next is the shape rolled from this billet. Following this the part is further shaped to desired contour by flame-cutting on a multiple-torch machine. Finally the piece is rough machined to required dimensions. Welding kerfs are provided in the same machining operation.

Another typical criterion is the welding conditions imposed by the design. Full-strength welds must be accessible so that they can be executed properly. Too, they must be designed

so that they can be inspected readily. The first question here for the designer is the quality of weld required.

As an example, *Fig.* 8 shows a typical subassembly with the welding completed as shown, which is accessible from the standpoint of both welding and inspection. Shrinkage and warpage have been eliminated as effects on the final assembly, and the other portion of the final structure also has been welded under most accessible conditions. It has been inspected and controlled separately from the standpoint of shrinkage or warpage. *Fig.* 9 shows this subassembly in its final position with the joining welds between the subassembly and the remainder of the structure executed through the openings.

Basic Component Is Hot-Rolled Plate

Definite trends emerge in the type and form of components being used in weldments. These trends are established naturally by economy and inherent design freedom. It is evident from the examples given that hot-rolled plate is the predominant component. Undoubtedly this is due mainly to the freedom in design that two-dimensional plates allow, for almost any conceivable size or shape can be designed using plate as the basic component.

Judicious use of other types of components in combination with plate provides added freedom and economy. The various methods of shaping or forming plate are utilized for the same reasons. The most widely used plate component is, of course, the flame-cut shape.

Bent plates typified by those shown in *Fig.* 10 are used to advantage in many applications. The stiffening side members of this hydraulic press frame are of plate formed on the press brake. Their use in this typical instance reduces the welding required, provides initial stiffness or stability in the component, and adds to the appearance of the weldment. The primary member of this frame is composed of plates formed on bending rolls and butt-welded.

Hot-flanged or "spun" heads often are used to advantage as components. In the weldment shown in *Fig.* 11, one head is halved to provide the vertical walls and rounded corners of the main portion of the structure. Welding is simplified and better appearance results. With the advent of quantity requirements there is a trend toward the use of hot pressings or stampings, as shown in *Fig.* 12. This one is approximately 53 inches x 10 inches x 6 inches, of 7/16-inch thickness. Two such pressings form the major portion of the housing pictured in *Fig.* 13.

Utilizing Drop-Forged Components

Designs based on quantity production include, to advantage, drop-forged components. They provide freedom of shape and eminent predictability where necessary, and forging die cost becomes insignificant in such quantity applications. *Fig.* 14 is an interesting specialized use of drop-forged components composed of several small forgings which are flash welded together. Design freedom in shaping for minimum weight and maximum predictability results.

Castings are utilized in a similar manner. Hundreds of tons of steel castings as shown in *Fig.* 17 are used as components in machinery weldments, and this use has opened a completely new market for steel castings as well as forgings. Generally such components are used in parts of a size or nature prohibitive or impossible as a one-piece steel casting or forging. *Fig.* 17 pictures a typical cast steel component. Clearly

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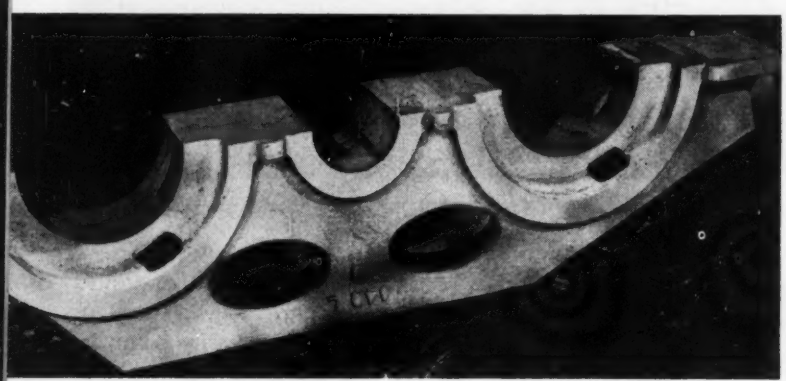

Fig. 17 — Left — This component is more satisfactory as a casting than as a built-up plate design. It is used in the weldment in Fig. 18

Fig. 18—Principal components of this frame are steel castings (Fig. 17) welded together to form a structure whose cost might be prohibitive cast in one piece

Applying Fatigue Test Data in Design

By Joseph Marin

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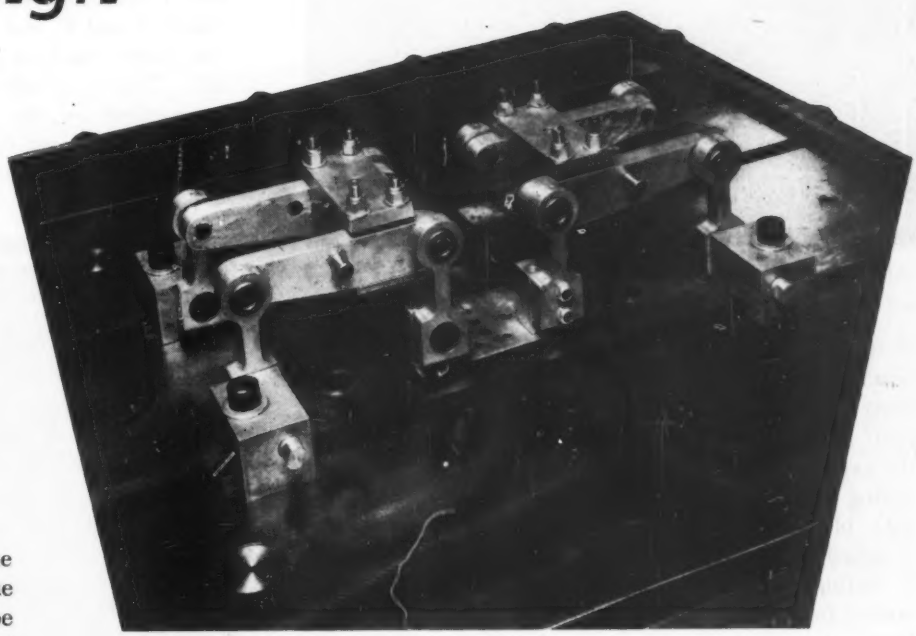


Fig. 1—Bending test setup in universal fatigue testing machine developed by Sonntag Scientific Corp.

IN THE design of some machine parts, consideration of the fatigue strength of the material to be used is highly important. In recent years the machine designer has used fatigue test data in determining the size of members subject to alternating stresses. The purpose of this article is to discuss the interpretation of these fatigue test results and to apply these results to the design of machine members.

Importance of alternating stresses in the design of machine and structural members depends upon the type of application considered. For example, in some structures the number of stress reversals are relatively small compared to those during the life of some machine parts such as railway axles, crankshafts and connecting rods of engines, automobile springs and chassis, lineshafts, gear teeth, valve springs, and turbine shafts.

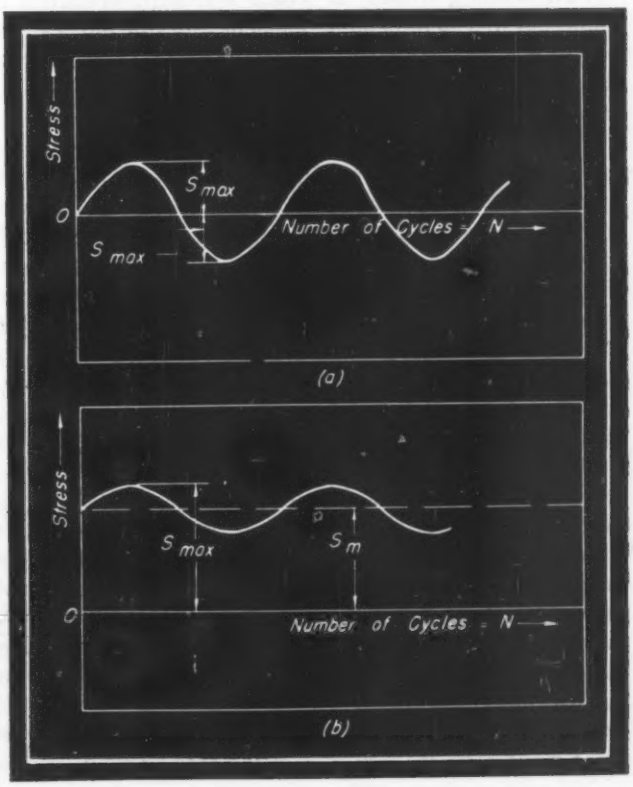
To determine the resistance of materials to fatigue stress conditions, three distinct kinds of fatigue testing have been developed; namely,

1. Testing of single crystals
2. Testing of specimens
3. Testing of machine or structural members.

Some work has been done on the first type dealing with single crystals (1)*. Although this work is of great value, the engineer must interpret the stresses as obtained by a stress analysis method in terms of the strength of the material. For this reason most of the work done on fatigue strength of materials has been of the second type, namely the testing of specimens, *Fig.* 1, under simple stresses such as bending, axial tension and compression, or torsion.

Lately the third type of fatigue testing has received considerable attention. Fatigue test results have recently been reported on machine and structural members such

Fig. 2—Variations in stress with time. At (a) are shown complete reversals while (b) indicates the more general condition of stress range met in most applications



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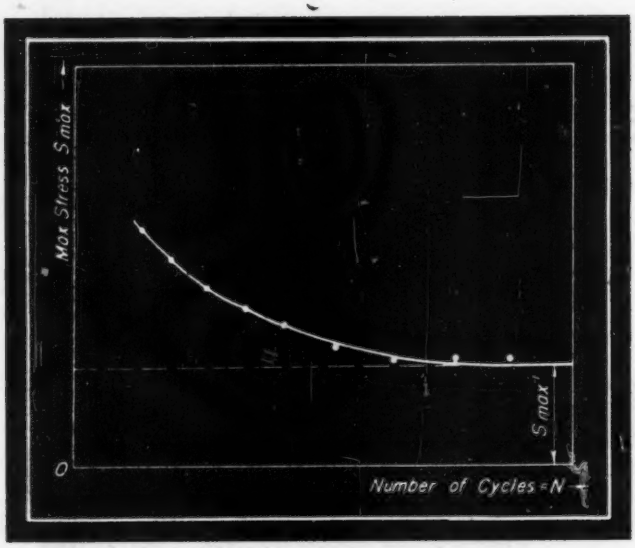


Fig. 3—Plotting of S-N diagram for a series of tests

as railway axles, riveted and welded joints, bridge rollers, beams, built-up columns, car wheels, springs, aeroplane parts and railway rails. Special testing machines have been built for testing complete members(2) but usually the expense or time involved does not permit this type of testing and the designer must then rely upon fatigue test data obtained by using standard size specimens and standard fatigue testing machines. The standard types of testing machines and test procedures used for determining fatigue strengths have been discussed in detail(3). Interpretation of the test data for purposes of design, however, has not received sufficient attention.

Interpretation of Fatigue Strength Data

Most fatigue-strength values are determined for reversed bending stresses in which a rotating specimen is subjected to a pure bending moment. In this test a stress which fluctuates from compressive stress to a tensile stress of equal magnitude is produced on the outer fiber, as shown in *Fig.* 2*a*. The machine designer, however, is interested in the more general case of stress fluctuation where the ratio between the maximum stress S_{max} to the mean stress S_m can be varied, *Fig.* 2*b*. An axial fatigue stress machine is used for this purpose. A series of specimens are tested with a constant value of the mean stress S_m and varying values of the maximum stress S_{max} for each test, *Fig.* 2*b*. In each test the number of cycles, N, to produce failure are determined and an S-N diagram is then plotted, as shown in *Fig.* 3. Other series of tests can then be made for various values of the mean stresses. From each S-N diagram the fatigue strength S_{max}' is selected. The basis for defining this strength may be either the stress for a specified number of stress cycles or the stress obtained by drawing the horizontal asymptote to the S-N curve, *Fig.* 3. In the determination of the fatigue strength there is one case, represented in *Fig.* 2*a*, in which the mean stress $S_m = 0$ and the stress is completely reversed from a minus value to an equal positive value. The value of S_{max}' for this condition of complete reversal will be designated by S_e.

Various methods of plotting have been used to represent graphically the influence of the stress ratio S_{max}'/S_m. *Fig.* 4 shows a method in which the maximum stress S_{max}' is plotted against the mean stress S_m. Point P in *Fig.* 4 then represents the results of one series of tests, as shown in *Fig.* 3. Point A is determined from tests refining the fatigue strength for complete stress reversal, since for complete stress reversal $S_m = 0$ and $S_{max}' = S_e$. Point C represents the case of failure by static tension, since for static tension $S_{max}' = S_y$ = the yield stress or stress defining failure by yielding and the mean stress $S_m = S_y$. Between these extreme stress conditions represented by points A and C, test data show that for ductile materials test points, such as point P, fall above the straight line AC. That is, a safe assumption is to consider failure as represented by the straight line AC. Recently J. O. Smith(4) assumed two straight lines in place of the line AC. Although Smith's interpretation represents the test

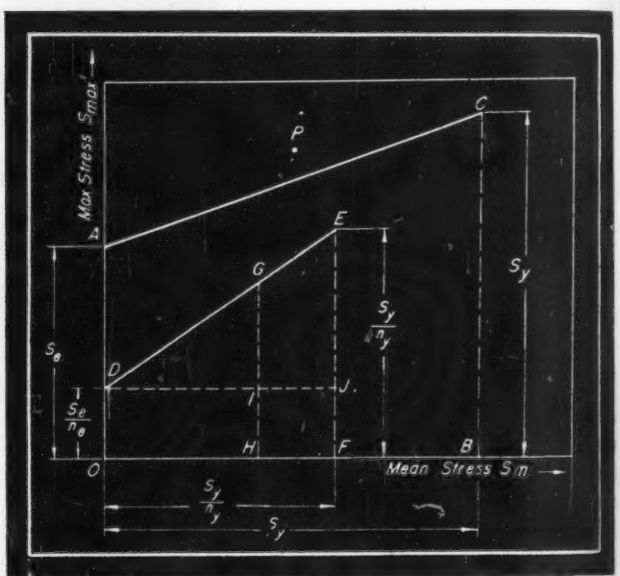


Fig. 4—Graphical representation of influence of stress ratio with maximum stress plotted against mean stress

data more accurately, the use of the straight line assumption given in *Fig.* 4 greatly simplifies application of the data in the design of simple stress members.

The design stress corresponding to the assumed failure line AC can now be determined. To do this let n_e = the factor of safety for dynamic or fatigue stress and n_y = the factor of safety for static stresses. Then $(OD) = S_e/n_e$ represents the design stress value for complete reversal of stresses and point E with coordinates $= S_y/n_y$ represents static tension. The ordinates to the straight line DE will then give the design or working stresses for intermediate ratios of the maximum to the mean stress. That is, the line DE represents the design stresses corresponding to the failure line AC. The equation for the line DE is determined by noting that the triangles DGI and DEJ are similar. It follows then that

$$\frac{GI}{EJ} = \frac{DI}{DJ} \text{ or } EJ \cdot DI = DJ \cdot GI$$

or

$$\left(\frac{S_y}{n_y} - \frac{S_e}{n_e}\right) S_m = \frac{S_y}{n_y}\left(S_{max} - \frac{S_e}{n_e}\right)$$

or, solving for S_{max} and simplifying,

$$S_{max}=S_m(1-kp)+\frac{S_e}{n_e} \quad \ldots\ldots(1)$$

where $p = (S_e/S_y)$ and $k = n_y/n_e$.

Since $S_e = pS_y = pn_yS_w$, where S_w = the static design stress for the material, Equation 1 can be written,

$$S_{max}=S_m(1-kp)+kpS_w \quad \ldots\ldots(2)$$

Equation 2 gives the value of the design tensile stress S_{max} for a given value of the mean stress S_m in terms of the material constants p, S_w and the factor of safety ratio k.

To represent the variation in the design stress graphically, Equation 2 may be divided by S_w, or

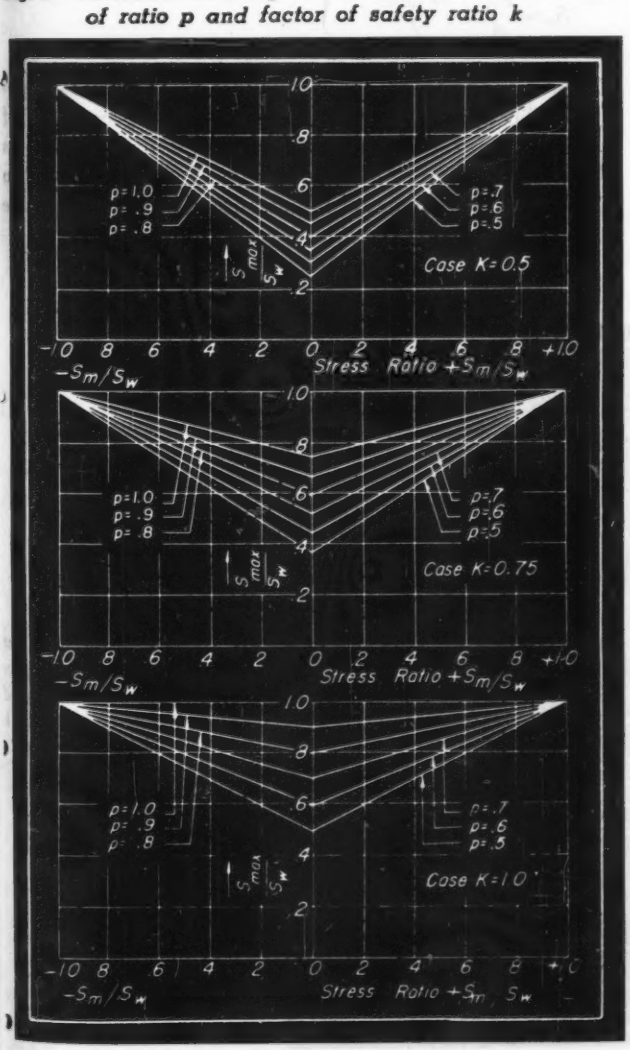
$$\frac{S_{max}}{S_w}=\frac{S_m}{S_w}(1-kp)+kp \quad \ldots\ldots(3)$$

or

$$R=s(1-kp)+kp \quad \ldots\ldots(4)$$

where R and s are the stress ratios S_{max}/S_w and S_m/S_w, respectively. The variation in the design stress ratio R is shown in *Fig.* 5 for various possible values of p and k.

Fig. 5—Variations in design stress ratio for various values of ratio p and factor of safety ratio k



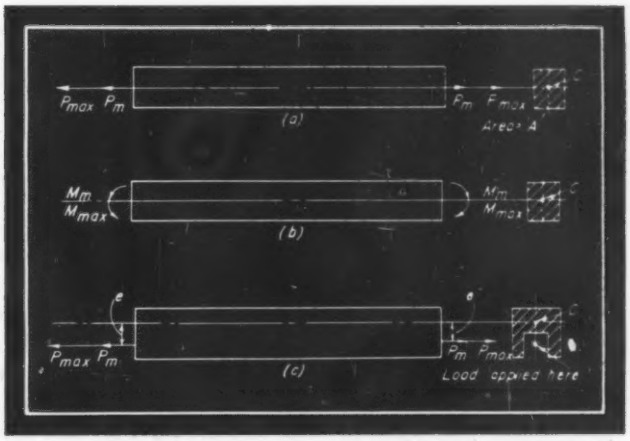


Fig. 6—Simple stress members having fluctuating axial, bending and eccentric fluctuating axial loads

Fig. 5 also shows values of the design stress ratios for cases in which the mean stress S_m is compressive; that is, for negative values of $s = S_m/S_w$. In plotting lines for the compressive values of S_m in *Fig.* 5 it is assumed that the strength values represented by line *AC* in *Fig.* 4 are the same for both tensile and compressive values of S_m. There are test data that show the foregoing assumption is on the safe side of the test results. Smith(4) gives some data for stress conditions where the mean stress is compressive to show that the equation $S_{max}' = S_m+S_e$ applies. For the design conditions the design stress would then be

$$\frac{S_{max}}{S_w}=\frac{S_m}{S_w}+pk \quad \ldots\ldots(5)$$

Although there is considerable test data to support the use of Equation 5, the use of Equation 3 for both tensile and compressive mean stresses seems justified in view of the simplification in using one equation in place of two.

The foregoing discussion has been confined to ductile materials. For brittle materials such as cast iron there is not sufficient information available from which to determine an empirical equation. Smith(4), however, on the basis of a few test results on cast iron, gives an empirical relation for the fatigue strength of cast iron. For tensile mean stresses this relation can be expressed by the equation

$$S_{max}'=S_m+S_e\left(\frac{1-S_m/S_u}{1+S_m/S_u}\right) \quad \ldots\ldots(6)$$

where S_u = the ultimate tensile strength.

Design of Simple Stress Members

Using Equation 2 it is a simple procedure to determine the cross-sectional dimensions of members subject to fluctuating axial, bending or combined axial and bending stresses, *Fig.* 6.

EXAMPLE I—AXIAL TENSION-COMPRESSION MEMBERS: Connecting rods and bolts are examples of straight members with constant cross sections subject to fluctuating axial stresses. Equation 2 can be used to determine the cross-sectional area A required for known values of the maximum (P_{max}) and mean (P_m) axial loads. Placing $S_{max} = P_{max}/A$ and $S_m = P_m/A$ in Equation 2,

$$\frac{P_{max}}{A}=\frac{P_m}{A}(1-kp)+kpS_w$$

or

$$A=\left(\frac{P_{max}}{S_w}\right)\left[\frac{1}{kp}-\frac{P_m}{P_{max}}\left(\frac{1}{kp}-1\right)\right]=\frac{P_{max}}{S_w}Q \quad (7)$$

The required area A is determined by Equation 7 for given loads P_{max} and P_m, material constants P and S_w, and factor of safety ratio k. The term Q in Equation 7 can be considered as a correction factor for dynamic loading since the first term P_{max}/S_w is the area required if P_{max} is statically applied. *Fig.* 7 gives diagrams for determining the area A when values of the load ratio P_m/P_{max} and constants k and p are known. The ordinates in *Fig.* 7 are values of Q, so that the area equals the ordinate times P_{max}/S_w. Values of the area can be obtained for various values of the load ratio P_m/P_{max} and for a number of possible values of k and p.

EXAMPLE 2—MEMBERS SUBJECT TO FLUCTUATING BENDING STRESSES: Many machine members are subject to stresses that are produced primarily by fluctuating bending moments. Leaf springs and some connecting rods are examples of such members. If the moments in these members vary from a maximum value of M_{max} to a mean value M_m, *Fig.* 6*b*, the maximum and mean stresses are:

$$S_{max}=\frac{M_{max}c}{I} \text{ and } S_m=\frac{M_mc} \quad (8)$$

where I = the moment of inertia of the cross section and c = the distance from the neutral or centroidal axis to the outer fiber of the member. Placing values of S_{max} and S_m from Equation 8 in Equation 2 and solving for I/c,

$$\frac{I}{c}=\left(\frac{M_{max}}{S_w}\right)\left[\frac{1}{kp}-\left(\frac{M_m}{M_{max}}\right)\left(\frac{1}{kp}-1\right)\right] \quad (9)$$

Equation 9 defines values of the section modulus I/c in terms of the load ratio M_m/M_{max}. Placing $I = Ar^2$ where r = the radius of gyration, Equation 9 can be used to determine the area A. That is, Equation 9 becomes

$$A=\left(\frac{c}{r^2}\right)\left(\frac{M_{max}}{S_w}\right)\left[\frac{1}{kp}-\frac{M_m}{M_{max}}\left(\frac{1}{kp}-1\right)\right] \quad (10)$$

The curves in *Fig.* 7 can be used for determining the required section modulus I/c provided the abscissas are considered to be the moment ratios M_m/M_{max} in place of the load ratio P_m/P_{max}. Then the ordinate multiplied by M_{max}/S_w equals the section modulus required.

Eccentric Loading Produces Bending

EXAMPLE 3—MEMBER SUBJECT TO ECCENTRIC FLUCTUATING LOAD: Members are sometimes subject to fluctuating loads that do not act along the neutral or centroidal axis. A bending moment is then produced as shown in *Fig.* 6*c*. If the applied eccentric load varies from P_{max} to P_m and acts with an eccentricity of e, the maximum and mean values of the stresses are expressed by the following:

$$\left.\begin{aligned} S_{max}&=\frac{P_{max}}{A}+\frac{(P_{max}e)c}{I}\\ S_m&=\frac{P_m}{A}+\frac{(P_me)c}{I}\end{aligned}\right\} \quad (11)$$

Equation 11 give the correct values of the stresses provided the cross section of the member has one axis of symmetry and the eccentric load acts on this axis as shown in *Fig.* 6*c*. Placing values of the stresses from Equation 11 in Equation 2 and noting that $I = Ar^2$, the cross-sectional area required is

(*Continued on Page* 200)

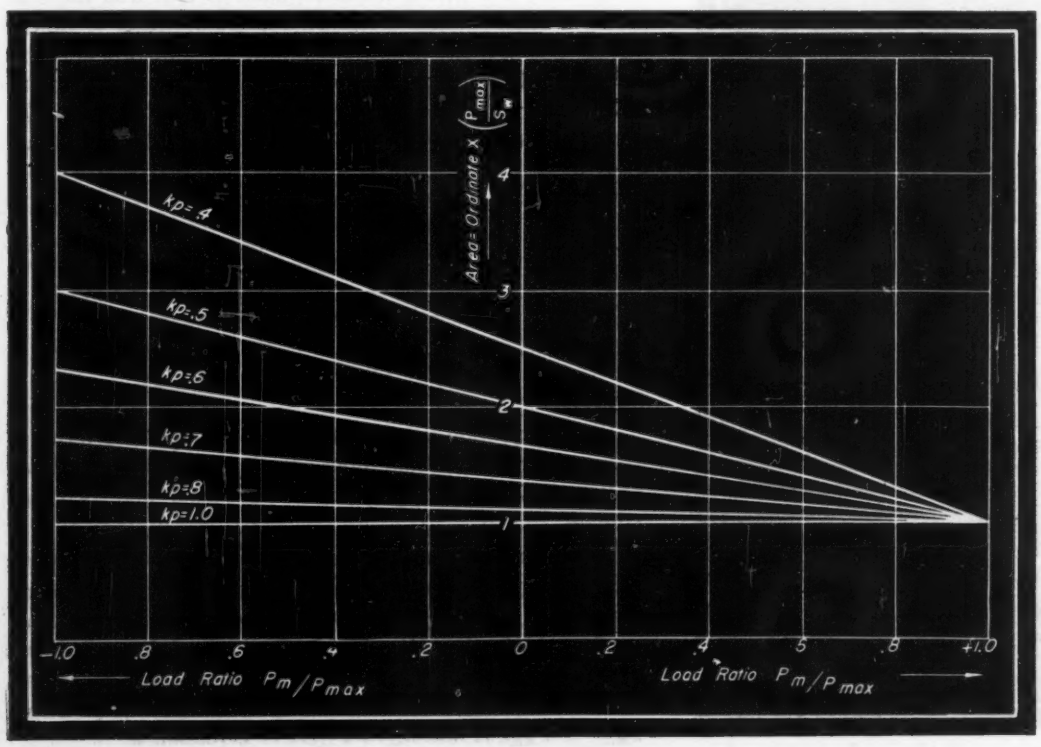
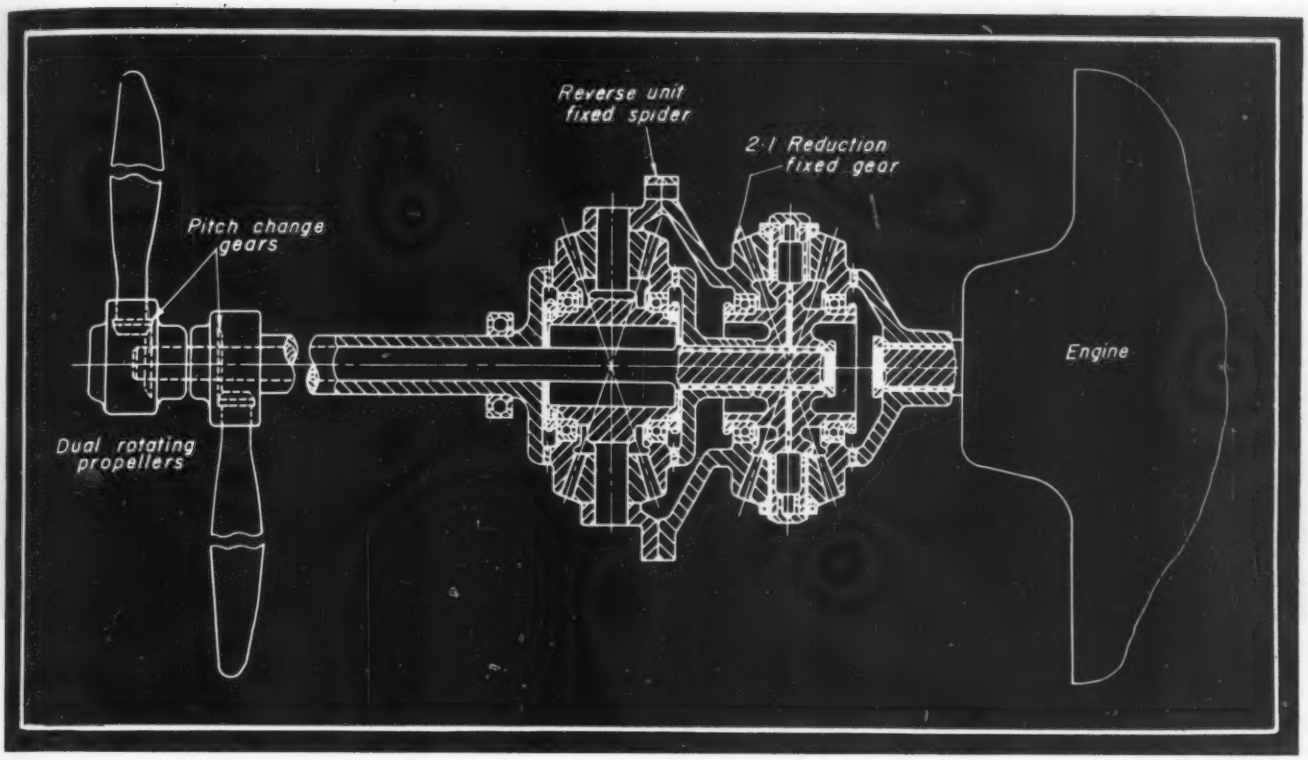


Fig. 7—Diagrams for determining cross-sectional areas when load ratio and constants k and p are known



Bevel Gears as Applied in Lightweight Designs

By L. J. O'Brien
Gleason Works

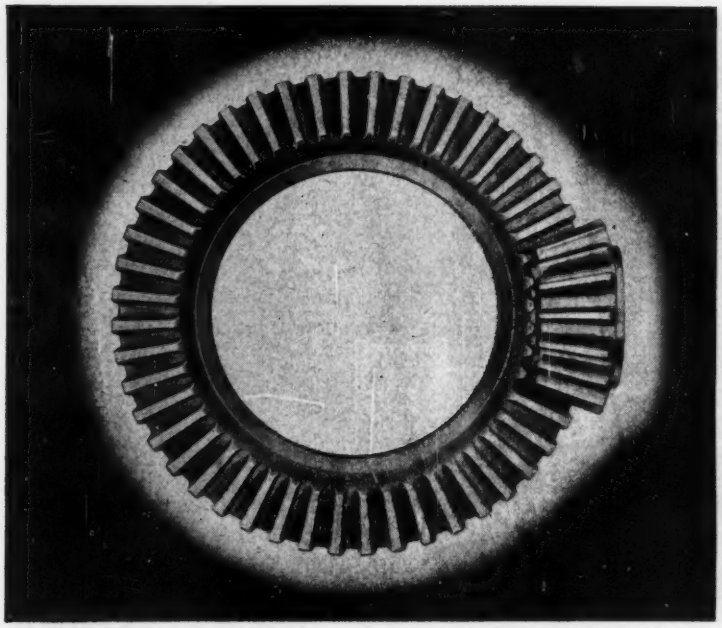
WITH the development and improvement in aircraft, demand for bevel gears to meet many special requirements has increased considerably. In power-operated control drives for retractable landing gears, bomb-bay doors, rudders, ailerons, etc., bevel gears in connection with hydraulic or electric motors have provided a logical system of supplying power to any location in the airplane. Bevel gears often permit the locating of accessories more conveniently and compactly around the aircraft engine. They maintain the proper relationship of all blades in variable-pitch propellers and provide a means of reversing shafts for dual-rotation propellers, *Fig.* 1. On fighting ships, bevel gears are used in the gun turrets, bomb sights, automatic pilots, etc. The main reduction in both airplanes and helicopters, as well as arrangements where the engine is located in the fuselage with a drive to propellers in the wings, are only a few of the other applications of bevel gears in aircraft.

Bevel gears differ in respect to the type of teeth used. The straight bevel gear, *Fig.* 2, has teeth which are straight and which, if extended inward, would intersect the axis. This is the oldest and most familiar form of bevel gear.

Illustrated in *Fig.* 3, the Zerol gear has been extensively used in aircraft to overcome some of the limitations of the straight bevel. This type of gear has curved teeth which are cut with face-mill cutters having multiple blades and can be ground, whereas machines for grinding straight teeth in bevel gears are not available. The curved tooth

Fig. 1—Above—Main drive from engine to dual-rotation propeller includes a reduction unit and a reversing unit, both employing bevel gears. Pitch-changing mechanism also uses bevel gears

Abstract of a paper presented at the recent national aeronautic meeting of the Society of Automotive Engineers in Los Angeles



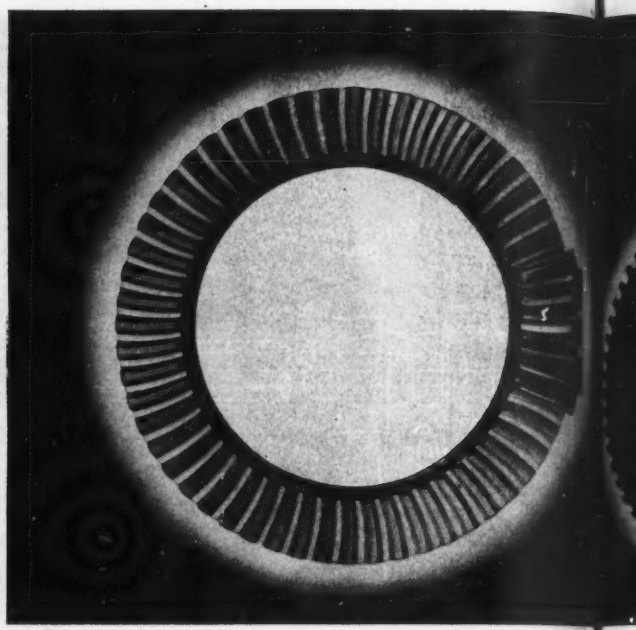


Fig. 2—Above—Straight bevel gear has straight teeth which, if extended inward, would intersect the axis

Fig. 3—Above, Right—On curved-tooth gear the length of tooth bearing can be controlled to allow for deflection

always has the advantage that the length of tooth bearing can be controlled by varying cutter radii so as to allow for different deflections of the gears under different loads. The curved-tooth gear has the same thrust and tooth action as a straight bevel and may be used in the same mountings. However, in curved-tooth gears all front and rear hub projections above the root line must be eliminated to provide cutter clearance.

Spiral-bevel gears as shown in *Fig.* 4 have curved, oblique teeth in which contact begins gradually and continues smoothly from end to end, consequently such gears are used for transmitting heavy loads at high speeds, especially where smoothness and quietness are required. Properly designed, they will have at least two teeth in contact at all times and for this reason will transmit as much as 30 per cent more power than the corresponding straight or curved-tooth bevels. The direction of endwise thrust in spiral teeth may vary with the direction of rotation. This has led to a misconception that spiral bevels are good for one direction of rotation only. The spiral-bevel rear axle on your automobile, however, will run equally well in either direction. The only requirement of a spiral-bevel gear set for reversible operation is that the members be held against thrust in both directions. No distinction is made between the efficiency of straight and spiral-bevel gears. Accurately manufactured gears of this type, properly adjusted in the mountings, are customarily rated at 98 per cent efficiency.

TABLE I

Maximum Design Loads for Aircraft Applications*

(Pounds per inch of face)

Teeth in Pinion	Diametral Pitch					
	3	4	5	6	8	10
12-14	2450	2120	1900	1730	1500	1340
15-18	2740	2370	2120	1935	1675	1500
19-23	3085	2670	2390	2180	1890	1690
24 and up.....	3465	3000	2685	2450	2120	1900

*Based on surface durability for case-hardened ground-tooth spiral bevel gears.

Note: Mountings and lubrication must be in accordance with best practice in order to transmit the above loads satisfactorily.

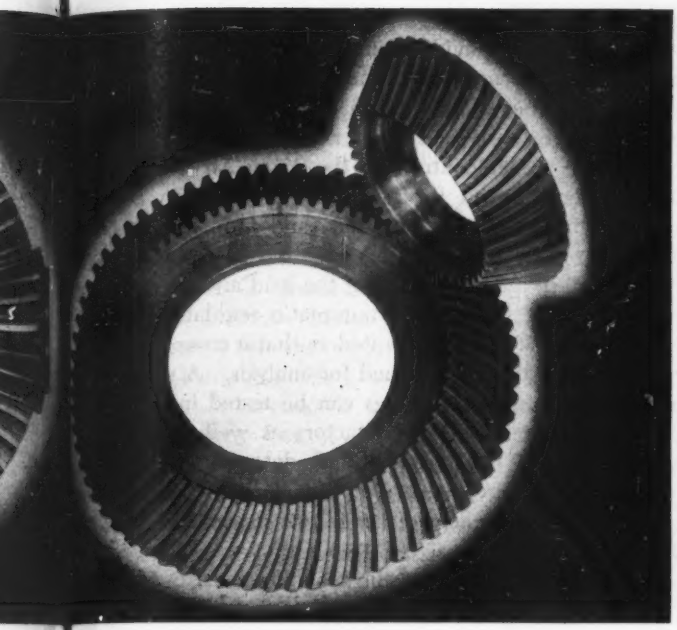
The hypoid gear and pinion shown in *Fig.* 5 are similar in general to spiral-bevel gears except that the pinion axis is offset from the gear axis. The gear and pinion shafts of the hypoid pair may continue past each other to allow compact multiple-drive arrangements.

Factors Affecting Bevel Gear Selection

First step in the design of gears for a particular application is to study all of the conditions under which the drive must operate. The type of gear to be used should be based on a consideration of speed, power, type of mounting, and requirement of operation. Where conditions are such that either a straight bevel or Zerol could be used, it is desirable to make the design such that the latter type can be incorporated without change. There have been several instances recently where production problems could have been simplified by incorporating ground Zerols in place of straight bevels. However, major changes in the original design would have been required to remove hub projections which prevented the cutting of the curved teeth.

Spiral bevels are strongly recommended where pitch-line peripheral speeds exceed 1000 feet per minute. In many applications, spiral bevels may be used to advantage at lower speeds because of greater smoothness and quietness. It is also well to remember that spiral bevels will, in most cases, provide a lighter overall construction due to their greater load-carrying capacity.

Practice for the last few years in aircraft work has been to select an initial gear size on the basis of the surface durability in pounds per inch of face. The maximum allowable load values for surface durability of ground spiral-bevel gears as used in aircraft are shown in TABLE I. For curved-tooth gears the allowable loads are taken as one-half those for spiral bevels.



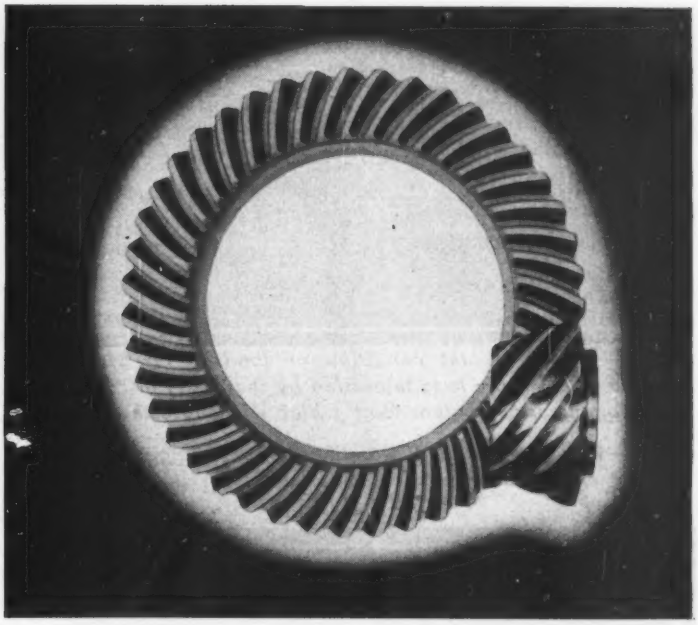


Fig. 4—Above, Left—Spiral-bevel gear has curved oblique teeth at least two of which are in contact at all times

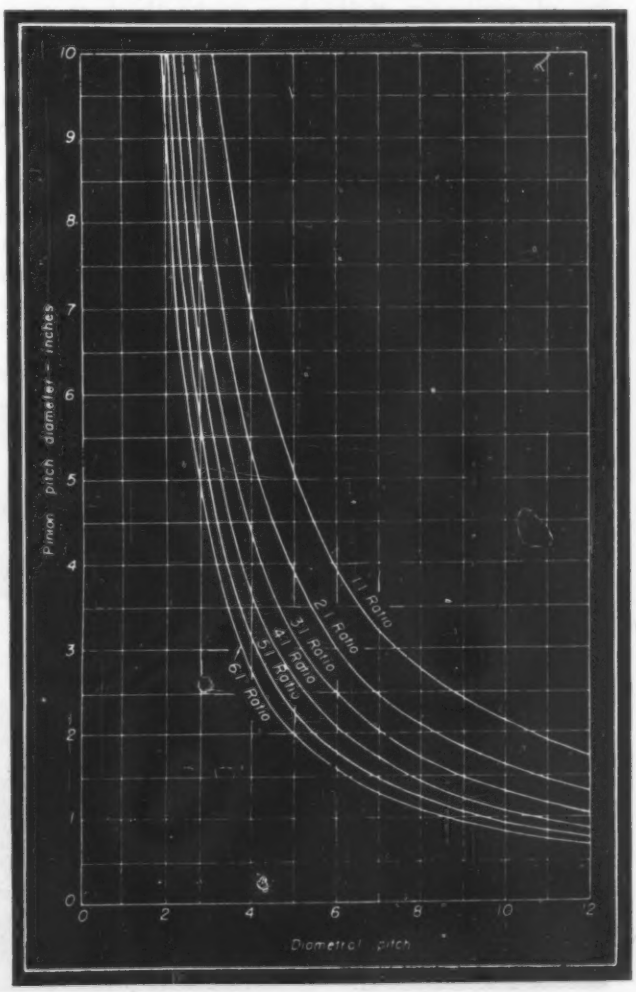
Fig. 5—Above—Hypoid gear has curved teeth like the spiral bevel, but pinion axis is offset from gear axis

Experience shows that the effects of the rigidity of the mounting and of lubrication cannot be too highly emphasized. The loads in TABLE I cannot be carried satisfactorily unless proper mountings and lubrication are provided. Because of this it is advisable to base the loads with respect to surface wear on mounting conditions, rather than on an estimate of theoretical surface pressure.

For good design, the face width should not exceed .3 of the cone distance for spiral bevels or .25 of the cone distance for curved-tooth bevels. No installation can be made absolutely free from deflections, and should these deflections be of a nature to concentrate the load at the small ends of the teeth, the entire load will be carried at this weakest point. Accordingly, failure will be more likely to occur on a gear with a longer face width than on one with a shorter face width and correspondingly coarser pitch at the small ends.

Taking the gear ratio desired and the approximate gear size determined as in the foregoing, a tentative diametral pitch can be selected from the chart shown in *Fig.* 6. It is of no importance in bevel gears to select an even or standard number for diametral pitch because with the method of cutting used, tools or cutters are not narrowly restricted to certain pitches. For instance, the same cutter may be used satisfactorily for either 4 or 4.17 D.P.

Fig. 6—Below—Chart enables designer to select tentative diametral pitch for 90-degree spiral-bevel gears



Checking Tooth Strength

Having determined the approximate gear combination, the next step is to check for tooth strength and fatigue life. To obtain the stress in a bevel gear tooth we have for a number of years used a modification of the familiar Lewis method. In the old Lewis method, the full load is applied to the tip of the tooth. In the modification of this method, the load is taken in the position when it is carried on one tooth. Thus, in *Fig.* 7 the pinion tooth at the left is just on the point of engagement so that the tooth at the right carries the full load. This is the worst condition of loading on the pinion tooth, since further movement to the right brings another pair of teeth into contact to share the load, while further movement to the left brings the

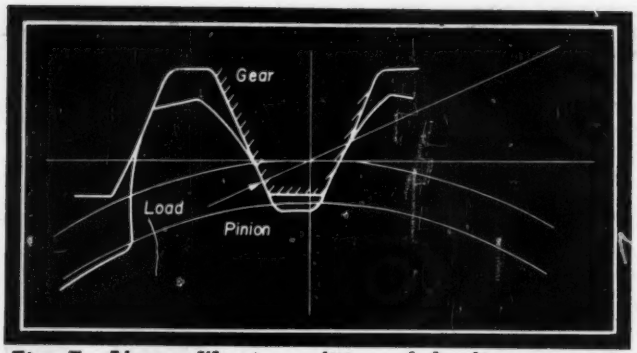
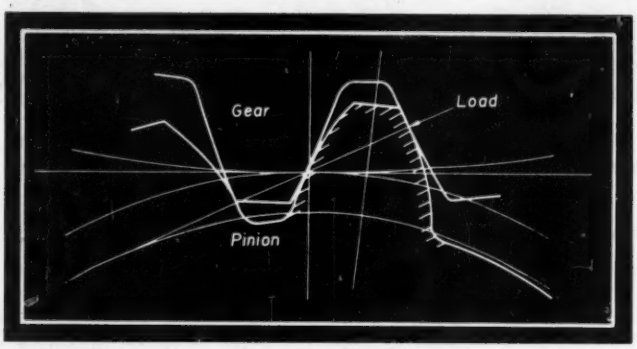


Fig. 7—Above—Worst condition of loading on pinion tooth is when whole load is carried by that tooth just prior to engagement of another tooth which will share the load

Fig. 8—Below—Position of worst loading on one gear tooth is established in same manner as for pinion



line of application of the force nearer the base of the tooth. Similarly *Fig.* 8 shows the worst condition of loading on the gear.

If the tentative gear size does not meet the requirements for fatigue life and static strength, a larger gear or coarser pitch combination will, of course, have to be used.

In 1931 the General Motors research laboratories published a stress cycle graph based on tests of a large number of rear-axle gears, with the stress estimated by the method described. This chart or graph has since furnished the background for predicting the life of a bevel gear set with reasonable accuracy. Numerous other tests being made are providing similar data for aircraft applications. The form of this chart is shown in *Fig.* 9.

For a number of months the Gleason Works has been conducting extensive tests on right-angle ground spiral-bevel gears under contract to the Army Air Corps. The machine constructed for this work is shown in *Fig.* 10. It is designed to test load capacity and endurance of bevel gears at speeds and loads encountered in aircraft drives. Provisions are made for varying the load and speed over a wide range of values, and automatic regulating and recording devices are incorporated so that a complete record of operational data is obtained for analysis. A wide range of bevel gear sizes and types can be tested in order that the effect of various design factors as well as operating conditions may be studied. The data obtained over a long-range testing program on this machine should add considerably to the field of bevel gear design and application.

General arrangement of the machine consists of a gear

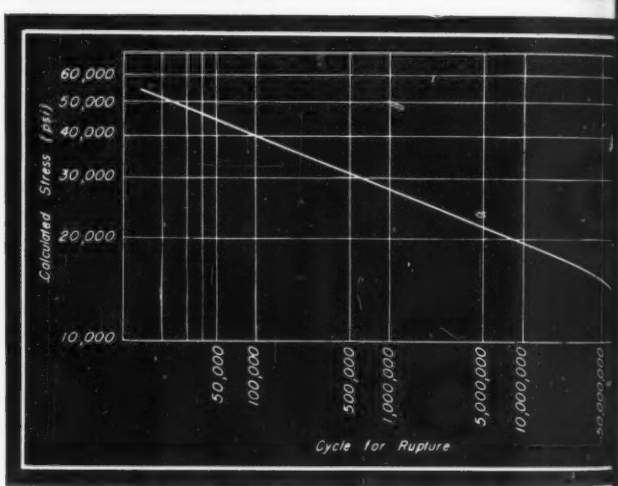
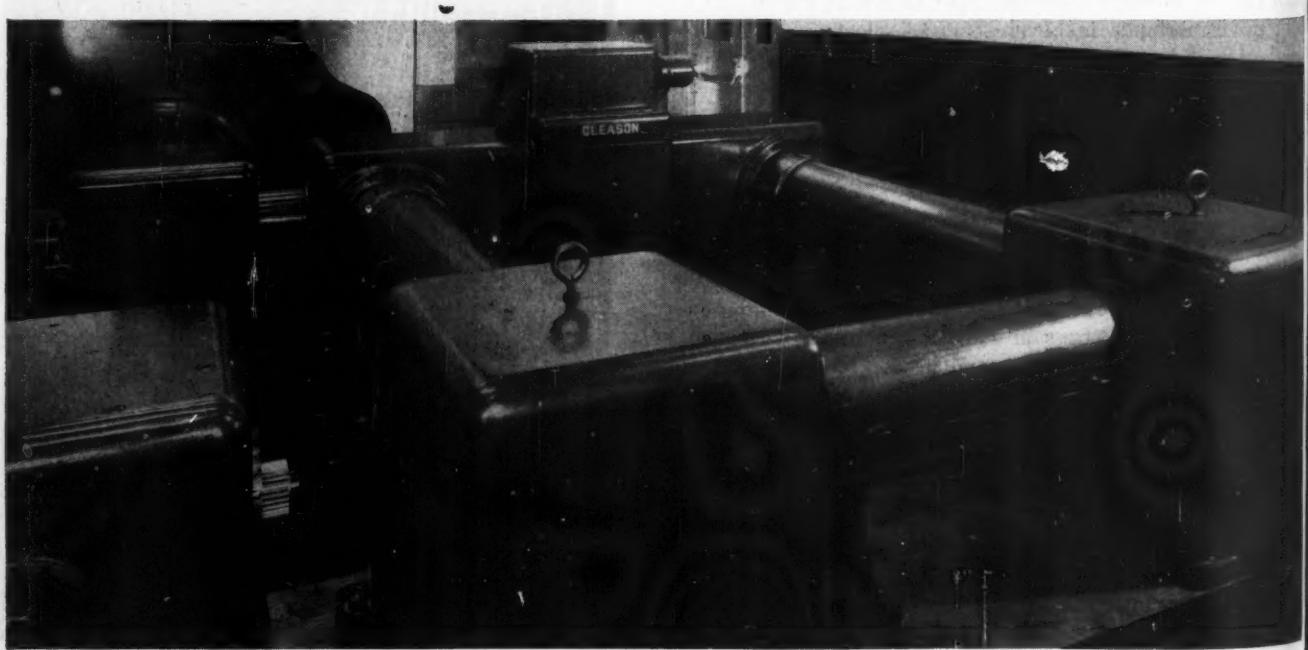


Fig. 9—Chart relating gear stress and gear life forms the basis for predicting life of a bevel gear set

Fig. 10—Machine for testing aircraft bevel gears has provision for applying loads up to 4000 horsepower



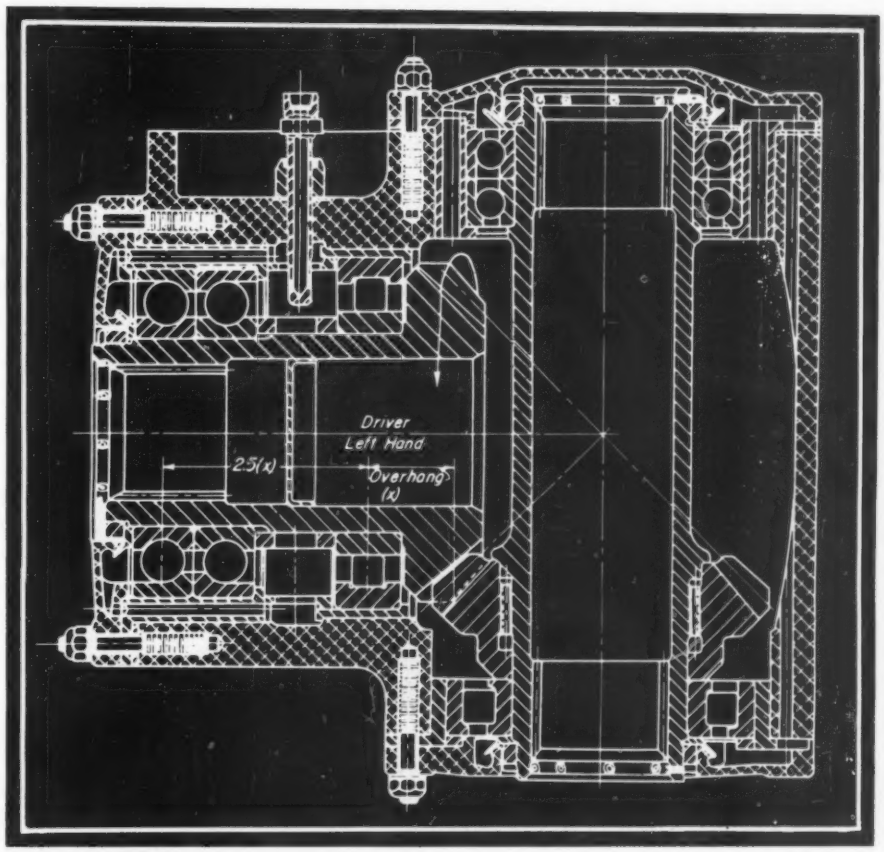


Fig. 11—Typical bevel gear mounting illustrates design which provides adequate support for the teeth

train arranged in the form of a square, making a closed mechanical torque system. Mounted at two corners of the square are the test boxes containing the gears being tested. It is possible to substitute a dummy box at one corner, permitting the testing of a single box if desired. At the other two corners are gearboxes to complete the square. Between these two boxes is a bevel gear mechanism by means of which the torque in the test system can be wound up and varied while running. The complete system is driven by an electric motor through one of two bevel gearbox assemblies which, with slight adjustment of the motor speed, will drive the pinion of the test box at any speed between 2400 and 3600 revolutions per minute. Loads up to 4000 horsepower at the speeds mentioned can be applied. The torque reaction of the pinion cage in the windup box is measured on a special scale arrangement. Permanently connected instruments record the torque, time and date at regular predetermined intervals throughout a test.

The operator has complete control of the machine and auxiliary apparatus from the control panel. Test information automatically recorded on charts or tapes consists of the following: Oil inlet and outlet temperatures, gallons per minute of oil flowing to the test box, speed of pinion shaft, and torque load as recorded by the scale mechanism. Multiple-point potentiometers record the temperatures of various parts of the boxes under test.

New Design Data Forthcoming

This machine has been described in detail to give a better picture of how the various factors affecting gear life may be judged. In the near future we hope to be able to publish design data based on the results of these tests, together with improved tabulated values of tooth form factors for strength of bevel gear teeth.

Tooth loads in spiral-bevel and hypoid gears produce a thrust along their axes, the direction of which depends upon the hand of spiral, the direction of rotation, and whether the gear is the driving or driven member. Bearing design must include proper provision to handle this thrust. In spiral-bevel and hypoid gears, the hand of spiral is denoted by the direction in which the teeth curve; thus, left-hand teeth incline away from the axis in a counterclockwise direction when an observer looks at the face of the gear, and right-hand teeth incline in a clockwise direction.

Thrust loads may be calculated for all types of bevel gears with the formulas shown in bearing and engineering handbooks. Often the mounting conditions will dictate the hand of spiral. However, where possible, the hand of spiral should be selected so that the axial thrust will tend to move both gear and pinion out of mesh or, at least, tend to move the pinion member out of mesh.

Gear shafts should be supported by bearings placed so as to provide the best support for the gear teeth. A typical right-angle gearbox containing a pair of miter gears is shown in *Fig.* 11. With the type of mounting used on the driver in this box, the overhang of the gear should be held to a minimum, and it is a good rule to spread the bearings at least 2.5 times the overhang. Note that the thrust bearings of the straddle-mounted driven member are at the opposite side of the gearbox from the gear. It is better to place the thrust bearings near the gear to eliminate the effect of housing expansion.

The gearcase should have sufficient rigidity to hold deflection and endwise displacement of the gears under operating loads to a minimum. A study of the various loads on the gears will suggest locations for ribs to reduce local bending in the case and provide a light but rigid construction.

Correct setting or adjustment of the gears at assembly is important. Provision should be made so that the pinion can be set to correct mounting distance by means of a simple gage at assembly. After the pinion is correctly positioned, the gear can then be adjusted to mesh with the mate in the position which gives the desired amount of backlash. A study of the tolerances of the component parts of the assembly should be made to determine their effect on the mounting distances of the gear parts. Shims of sufficient thickness should then be provided in the

design to permit adjustment of gears to correct position.

A means of inspecting the gears in mesh is desirable both from an assembly standpoint and for periodic check. An inspection hole and cover should be arranged so that the contact pattern can be observed on the teeth of both members of the gear set.

Problems in machining, gear cutting, inspection, and assembling can often be simplified by the proper design of the gear blanks. It is desirable to locate the gears in manufacturing operations on the same surfaces from which they will be mounted in assembly. Care should

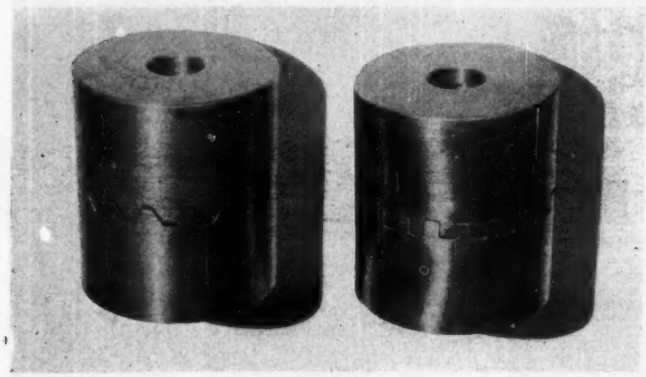


Fig. 12—Curvic splines are outgrowth of developments on curved-tooth bevel-gear cutting and are used in a variety of applications, including couplings and clutches

be taken, therefore, that the mounting surfaces are of generous proportions and placed where they can be used for locating in cutting the teeth, etc. Spline-bored gears require a cylindrical centering fit to insure maintenance of concentricity of the splines with the gear teeth. On gears in which the splines must be hardened, it is recommended practice to take this fit on the minor diameter of the splines, the major diameter to have clearance. Another method is to provide a suitable length of cylindrical fit at both ends of the splines and to use the splines only for driving, as illustrated by the driven member in *Fig.* 11. Since heat distortion may introduce both twist and out-of-round conditions in the splines, which cannot be corrected, it is important that the splines be of no greater length than required for load transmission.

A sufficient amount of metal should be provided under the gear teeth to give proper support. The general rule is that the rim thickness under the teeth should not be less than the whole depth of the tooth, at the ends of the teeth as well as at the middle.

Splash lubrication is generally satisfactory for peripheral speeds up to 2000 feet per minute. For higher speeds, some special provision should be made for directing or forcing oil into the mesh point.

Type and scope of gear drawings will vary with the procedure followed in a particular plant. A gear-specification drawing, containing all the dimensions which define the gear and those which are established by the relation of the gear to other parts, is usually required. These dimensions include the gear tooth specifications for proper operating contact with the mating gear. Specifications of material and heat treatment, and notes for inspection limits also should be given. The amount of information on the drawings is sufficient where both the gear and its mate are supplied by the same shop, because the responsibility for satisfactory tooth contact then lies with one maker. If, however, the gears are to be made in different shops, duplicate sample or master gears should be provided each shop to maintain interchangeability.

The method of dimensioning should definitely relate the crown point of the teeth to the mounting shoulder. Often drawings of gear blanks show dimensions in relationship to the cone center of the gear, causing confusion in the shop in turning the blanks. The cone center is a theoretical reference point in space and is of no value to the machinist in the manufacture of the blank. It is obvious that for practical reasons all of the dimensions of the blank should be such that they can be measured on the blank itself.

Tolerances of shafts, gears, etc., should be consistent and in accordance with the particular requirements of the job. It should also be remembered that working tolerances are required for the various operations involved in manufacture and that the final results cannot be better than these working tolerances. For example, a .001-inch tolerance on the bore may be combined with runout due to transfer of locating surfaces between operations or added to a series of tolerances used in each operation, and it would not be reasonable to expect .001-inch eccentricity of the pitch line in the final part. For those reasons, it is desirable to hold all blank tolerances to as close limits as possible throughout various manufacturing operations. Our experience in general has been that the best overall results are obtained with casehardened gears. It is a recognized fact that the casehardened section adds considerable strength to the gear tooth in addition to providing a good wearing surface.

Curvic Spline Couplings and Clutches

Although strictly speaking not a bevel gear, the curvic spline is a recent development which is an outgrowth of curved-tooth bevel-gear cutting. Two types are shown in *Fig.* 12. These splines are used in fixed couplings, semiuniversal couplings, overload clutches, and engaging-type clutches. The splines are cut and ground on standard bevel-gear cutting and grinding equipment, using standard cutters and grinding wheels. The teeth are curved and the amount of tooth contact can be controlled. Pressure angles from 0 to 30 degrees are used, depending on the application. Accuracy obtained in grinding these splines is such that alignment of parts may be maintained to extremely close tolerances.

One of the large radial engines now in production has a built-up crankshaft with parts coupled by ground curvic splines. Recently a two-throw crankshaft for a small auxiliary engine was built up from completely interchangeable parts. The crankpins, crankcheek and main-bearing members were hardened and individually finished to final size. Curvic face-splines were then ground from the solid in each part as the final operation. With this design a new crankpin or center member could be inserted without removing the crankshaft from the engine.

As yet, all the possibilities of this type of spline have not been fully developed. To date, in all applications which have demanded extreme accuracy and maximum load-carrying capacity, the use of these splines has been highly satisfactory and it is expected that this method of coupling will soon become an accepted standard.

MACHINE Editorial DESIGN

Military Research Must Go On!

IT has been said that if Germany could have started using its secret V-weapons six months earlier, England might have been blasted out of existence and the whole course of the war changed. Be that as it may, the fact remains that German research and development must be credited with having produced extraordinary weapons of war that have given and will continue to give lots of trouble.

By the same token, the research and development work that produced the Spitfire—and the part this type of plane played in the "Battle of Britain"—must be acknowledged, as also must be the research behind the development of the radar equipment that has been put to effective use on so many classes of military equipment by the Allies.

Many other instances of scientific research and development could be cited, but the foregoing undoubtedly will serve to emphasize the drastic need for this country to keep ahead in its programs and plans covering research work on military equipment. All activities along these lines obviously will be pursued with unremitting vigor to the end of the war, but they furthermore should be extended into the period after peace is signed. As Rear Admiral J. A. Furer told the Industrial Research Institute recently: "Americans are now ready to face the fact that the war to end all wars has not yet been fought . . . we must never again take the chances which we took after the last war!"

Peace brings forgetfulness, but we cannot afford—regardless of leagues, pacts or alliances—to slip back into the blind attitude of mind that overtook the nation when it was agreed that armaments be scrapped after World War I. Regardless of how successfully the present war is eventually brought to a conclusion, others seem more than likely to follow because of the tremendous upheaval in foreign countries and the trend toward changing philosophies and forms of government.

Extensive military research and development, both now and after the war, would go far as an insurance against future conflict. All possible support should be given to efforts in that direction, whether sponsored or controlled by industry or government, or both.

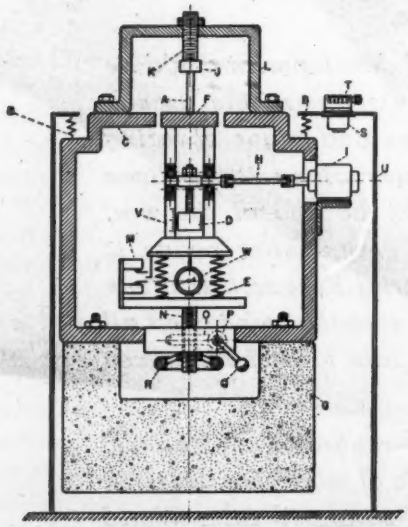
L. E. Jermy

Outstanding Designs

Fatigue Testing Machine

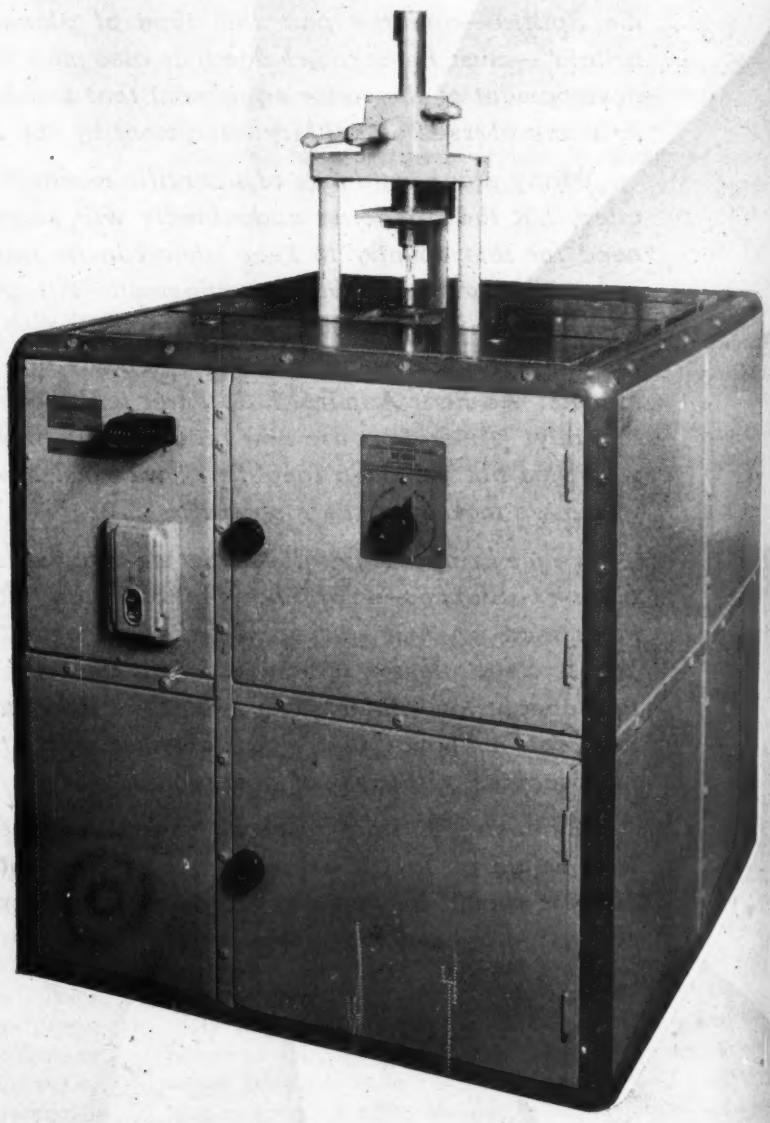
No gears, cranks or cams are employed in the mechanism of this fatigue testing machine developed by Sonntag Scientific Corporation. The fixed alternating force it develops is produced by a centrifugal-force type of mechanical oscillator which is direct driven from a synchronous motor through a flexible shaft.

Magnitude of the alternating force is adjustable by screw-type oscillator setting and the machine operates below resonance with helical springs utilized for preload and inertia force compensation. Mechanism for setting the preload springs is of screw type, worm driven, and employs handwheel for locking. Guides for the oscillating head permit vibration along one axis only, all other components being eliminated. Number of oscillatory cycles is indicated by a counter which registers in units of 100. An automatic shutoff is incorporated to stop the machine whenever a specimen fails. To eliminate transmission of vibration to the floor and surrounding areas, the main unit of the machine is suspended on helical tension springs from the outer frame. Tapped holes and T-slots are provided in both the reciprocating and fixed platens.



Legend

A Test specimen
B Mount for vibration isolation
C Frame
D Mechanical oscillator
E Springs for preload and inertia-force compensation
F Reciprocating platen
G Concrete stabilizing block
H Flexible drive shaft
I Synchronous electric motor
J Stationary platen
K Adjusting screw
L Holding fixture
M Microswitch cutoff
N Preload adjusting screw
O-P-Q Worm drive for preload adjustment
R Lockwheel for preload screw
S-T Cycle counter
U Sheet-metal housing
V Scale for indicating magnitude of oscillating force



Spot Welder

Upper and lower arms of this recently developed spot-welding machine manufactured by Sciaky Bros. are of hard-rolled solid copper, machined at one end only to accommodate water-cooled, ejector-type electrode holders in vertical position. A double-acting air cylinder, anchored to a bracket under the top cover of the machine, is connected to the rear end of the lever which holds the upper arm. This cylinder houses two pistons, one of which provides the operating stroke of 5/8-inch and the other a retraction stroke of 5 inches. Air to the cylinder is controlled by a solenoid-actuated valve.

The fulcrum point about which the upper arm pivots is equipped with heavy-duty, tapered-roller thrust bearings and is provided with means of adjustment to compensate for wear. Secondary sections of the transformer employed are of rolled electrolytic copper, water-cooled by means of copper tubing brazed to the periphery of the section. Transformer primary is tapped to give eight steps of heat regulation controllable by a switch at the side of the welder.

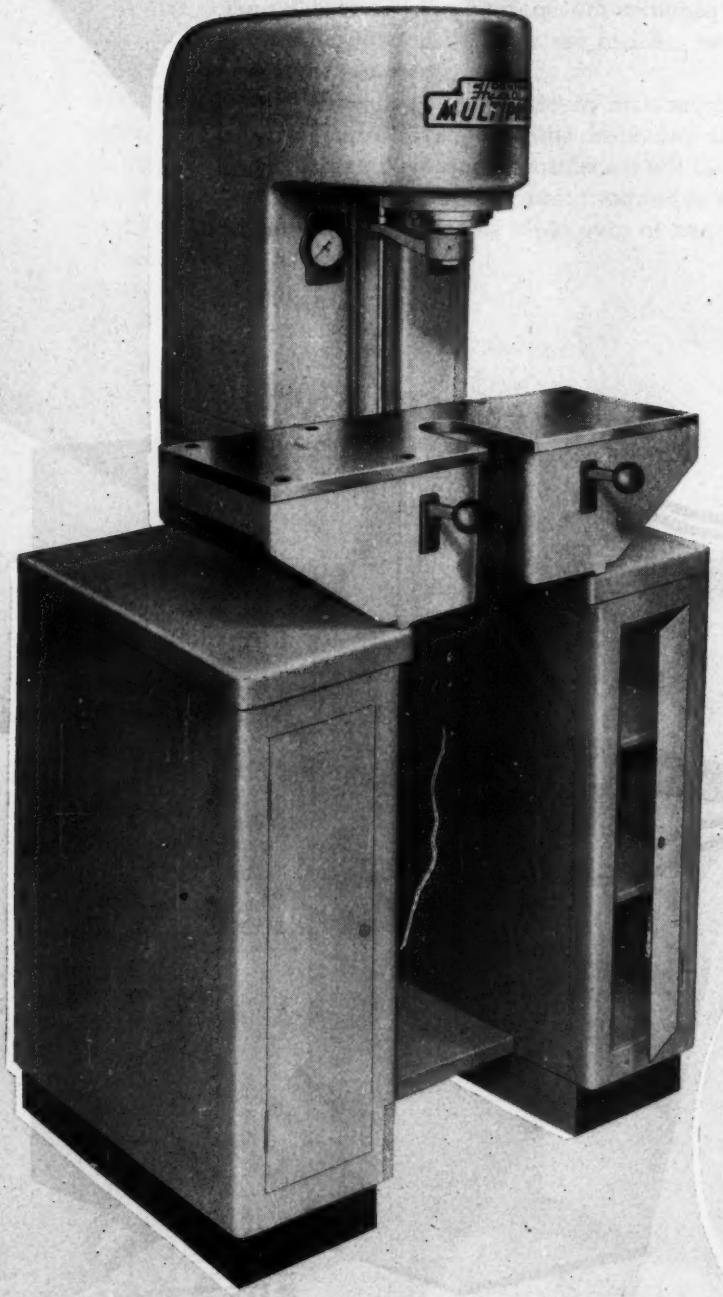
Two water circuits are employed for cooling, the transformer and electrodes being cooled by one circuit and the ignitron tubes by the other. Cooling coils are of copper, copper-nickel or nickel-copper, with hose connections where required. Frame and base are of welded steel construction and access to the interior is provided to permit making adjustments, servicing, etc.

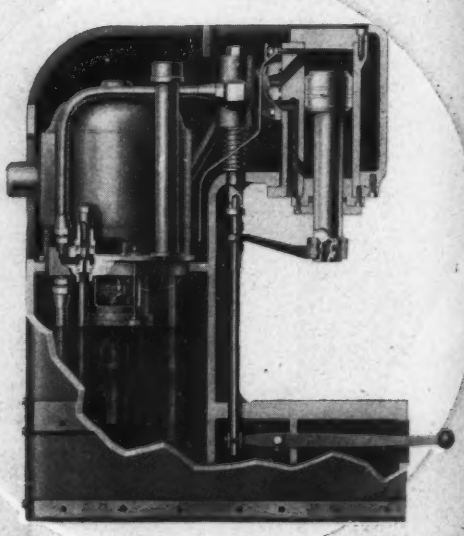
Hydraulic Press

In this machine the hydraulic pump, its electric motor and the pressure-relief valve are mounted as a unit assembly on the top cover of the oil reservoir which is formed by the machine base. Power cylinder and ram assembly is a double-acting unit capable of exerting 8000 pounds pressure downward and 5000 pounds upward. Cylinder and its bolted head are of high-grade cast iron and the one-piece ram and piston are of casehardened, ground steel. Steel piston rings form the piston-to-cylinder seal and chevron-type leather packing seals the opening at the lower end of the cylinder.

Located in the throat of the machine is the shipper rod. This, actuated by the two hand-control levers, operates the four-way valve which in turn controls movement of the power piston. An adjustable stop collar, mounted on the shipper rod, is contacted by the ram guide arm at the end of the upstroke to limit ram travel upward any distance between 1/16-inch and 6 inches. The control levers are balanced to permit finger-tip control.

Positioned at eye level on the press throat wall is a dial gage which indicates in tons and pounds the ram pressure being exerted at any given moment. To prevent wear during continuous operation, the gage is fitted with a shutoff valve. This modern, multipurpose press is manufactured by The Denison Engineering Co.

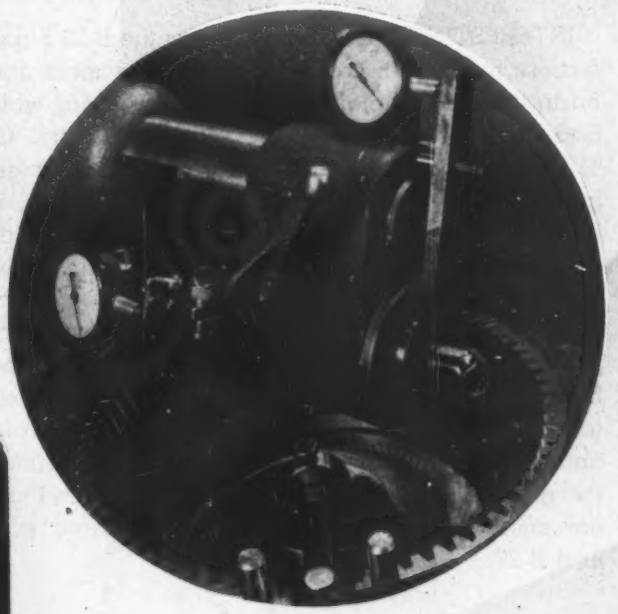




Internal Gear Checker

Principal elements of this machine manufactured by National Broach and Machine Co., are a rigid base, the top of which is finished, a work spindle mounted on precision bearings, an adjustable counterweighted column, and a checking head carried on the end of an overarm. The checking head is adjustable axially to and from the work-spindle axis to accommodate gears of various diameters.

The checking-head spindle, at right angles to the work spindle, is adjustably spring loaded to advance it toward the work-gear teeth and is retracted by means of a hand lever located at the front of the checking head. Carried at the forward end of this spindle is a rigid ball-pointed contact finger and a movable pivot finger. The latter actuates the plunger of an indicator on which errors in tooth spacing can be read in .0001-inch increments. At the rear end of the checking-head spindle is an arm which contacts the plunger of an indicator (.0005-inch increments) which is located on top of the checking head. This registers the axial movement of the checking-head spindle for measuring eccentricity errors of pitch and root diameters.



An adjustable gear-tooth locater mounted on the table positions the work gear with reference to the ball-point contact finger and prevents skipping a tooth when indexing the work gear.

Design Roundups

Represents Eight Million Hours

INTERESTING LIGHT was shed on the B-29 Superfortress recently in connection with a graphic demonstration of the plane's fire control system which had been developed by the General Electric Co. The plane now represents some 8,000,000 engineering hours, with engineering which normally would have taken five years compressed into two. More than 1000 engineers are working on continual improvement, incorporating the lessons of battle experience. Efficiency in production at the three prime plants which produce the Superfortress is indicated by the fact that while it required 157,000 man-hours to complete the first plane, it now requires only about 57,000. Something over 30,000 man-hours is the goal now in sight. Apart from rivets and similar small parts, there are 55,000 numbered parts in a B-29.

Gets Data Quicker

ENDURANCE TEST methods which previously required five or six years have been developed to obtain the necessary data within a month on high-temperature alloys used in steam turbines, jet propulsion units, etc. By ordinary means it takes years to stress an alloy sample held at 1500 degrees Fahr. a quarter of a billion times.

In the special machine the test piece is clamped in a furnace that maintains the desired temperature. Stress is applied at the rate of 120 times a second by two stator coils connected to a 2-phase, 60-cycle supply line. The entire system is mechanically tuned to resonance at this frequency to minimize the driving force necessary. When the first crack occurs the machine automatically stops through a control activated by the reduced stiffness of the specimen. Tests of high-temperature alloys up to a billion applications of stress have been made in 100 days.

Although no comparative data are available between five-year tests and accelerated tests, at elevated temperatures, data for room-temperature tests indicate there are no wide variations between the two types of endurance test at that temperature. In other words, the effect of recovery seems to be negligible. The accelerated tests, however, more nearly approach the frequency rates of the turbine blades and jet units mentioned and therefore more nearly approach operating conditions.

Enameling Becomes More Adaptable

SUCCESSFUL APPLICATION of white or pastel cover-coat enamel without prior application of a base or ground-coat enamel has long been sought. Solution now appears to have been found by Inland Steel Co. in the development of a new vitreous enameling alloy steel which contains titanium in a quantity dependent upon the amount of carbon in the base metal. The titanium eliminates pitting, black specking and blistering that occur during firing when enamels are applied to standard enameling stocks without application of a ground coat. Significant is the fact that a thinner covering of enamel produces a superior surface and one less liable to chip, as compared with the customary more costly multicoat job. Expectation is that the new alloy steel and process will expand use of vitreous enamel on many types of home equipment, such as refrigerators, washing machines, kitchen appliances and the like. Indications are that drawing properties are fully equal to the best deep-drawing iron and steel sheets, including carbon steel sheets furnished to the automotive industry. Also the new alloy does not age strain. Thus, shallow panels of large areas remain flat and true to shape after drawing.

Jet Engines Simplify Design

INHERENT SIMPLICITY of jet-propulsion engines greatly facilitates their production and operation. Inasmuch as they operate without appreciable vibration they may be tied direct to the air-frame. Relatively few gages and instruments face the pilot in the cockpit because there are no propeller or fuel mixture controls.

Tables and Charts Aid in Beam Vibration Problems

By C. R. Freberg and E. N. Kemler
School of Mechanical and Aeronautical Engineering
Purdue University

GENERAL solution of beam vibration problems is quite involved, but certain types of such problems that are frequently encountered can be solved simply and easily by using charts and tables. Perhaps the most common problem is that for beams such as used for general structural and mechanical purposes. The transverse vibration of long shafts, as well as many other machine elements, is in this classification. Another class includes control tubes, rods, and other links such as exist in all types of moving machines and aircraft control systems. Many other members used in machines or structures such as tension and compression or column members likewise often have to be checked for their vibration characteristics.

This data sheet covers the application of charts and tabulated values to the determination of the natural frequency or critical speeds of such parts when acting as simple uniform beam members and when the only load which they carry is their own weight. The natural frequency of beams fulfilling these requirements can be calculated from the equation

$$f_n = C\sqrt{\frac{gEI}{wL^4}} \quad \ldots\ldots (1)$$

where f_n = Natural frequency in cycles per second
E = Modulus of elasticity in pounds per square inch
I = Moment of inertia of section in inches^4
g = 386 inches per second per second
w = Distributed weight in pounds per inch
L = Beam length in inches

The value of C in Equation 1 will depend on the manner in which the beam is supported and the particular mode or critical speed at which the beam is vibrating. The actual calculation of C for each case is quite involved. Fortunately most cases have been worked out and are available in the literature (see References 1 and 2). *Fig.* 1 lists in tabular form various types of beams commonly encountered and the various modes of vibration up to the fifth critical. Each of these diagrams shows the mode of vibration and the value of C which applies. The position of each mode from the right-hand end, as a decimal fraction of the total beam length, also is shown. With this information it is possible to determine the various frequencies of the common type of beams.

Determining Method of Support

The data for simply supported beams also apply to the usual pin-ended beams or machine members. When the ends are welded or riveted, the ends may be assumed fixed. The fixed-hinged beam applies to elements where one end is fixed and the other is pin-connected. The free end beam would correspond to the vibration of a beam or rod lying on a floor or other support which did not transmit any moment or force to it in the plane of vibration. An airplane in the air or a ship in the water may be considered as representative of this type motion in a simplified analysis. The hinged-free beam would correspond to any member pinned at one end and free at the other. Rotating members of many types may fit this condition. Rotating wing aircraft with large horizontal rotor blades is one example of the hinged-free beam.

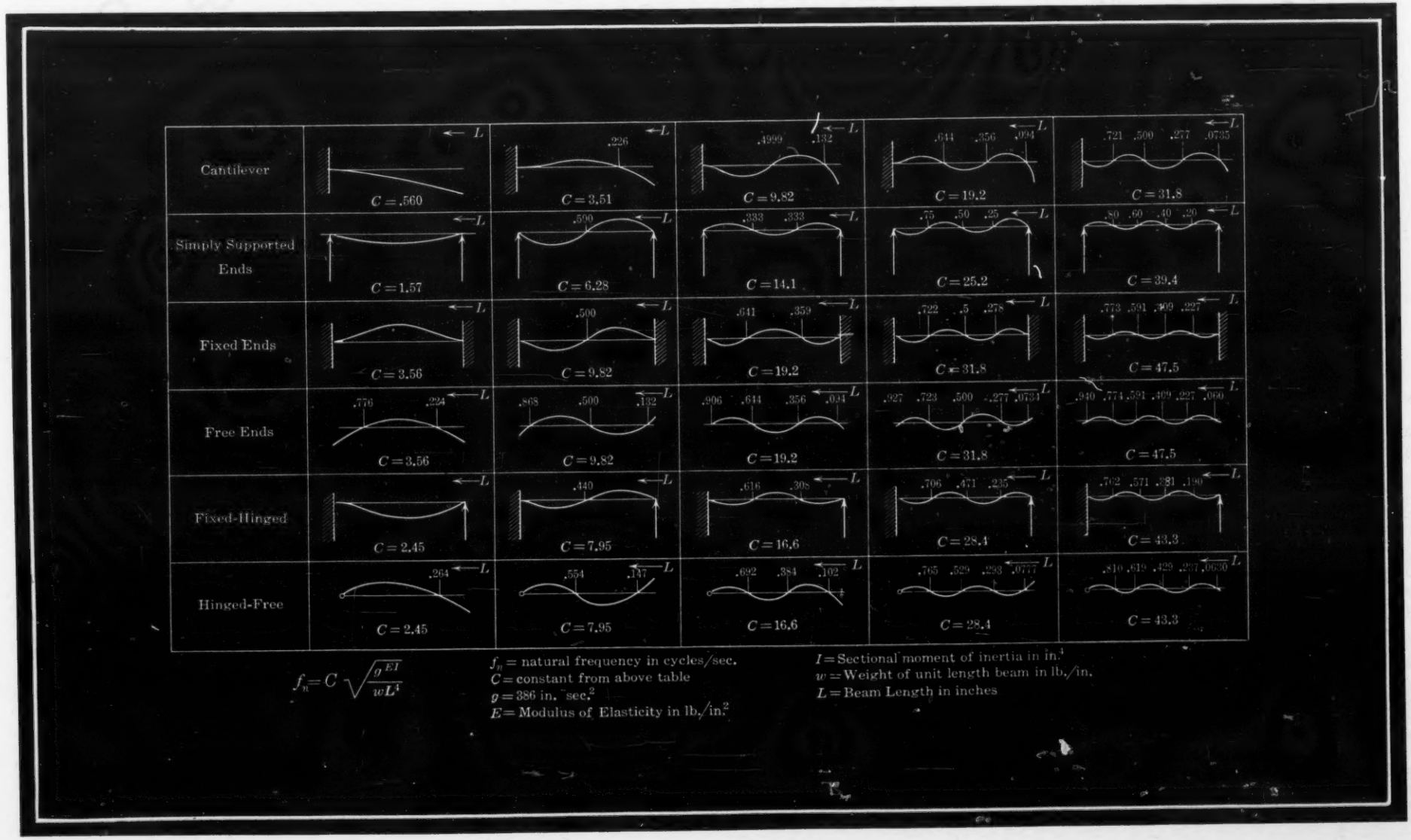
For a steel beam carrying its own weight Equation 1 reduces to

$$f = 1{,}190{,}000C\frac{\rho}{L^2}$$

(*Concluded on Page* 162)

Fig. 1—Below—Modes of vibration up to the fifth critical and the corresponding values of constant "C" are given for beams with various forms of support

Fig. 2—Opposite Page—Chart enables rapid determination of natural frequencies of beams when radius of gyration, support, and length of beam are known



$$f_n = C\sqrt{\frac{g\,EI}{wL^4}}$$

f_n = natural frequency in cycles/sec.
C = constant from above table
g = 386 in./sec.^{2}
E = Modulus of Elasticity in lb./in.^{2}
I = Sectional moment of inertia in in.^{4}
w = Weight of unit length beam in lb./in.
L = Beam Length in inches

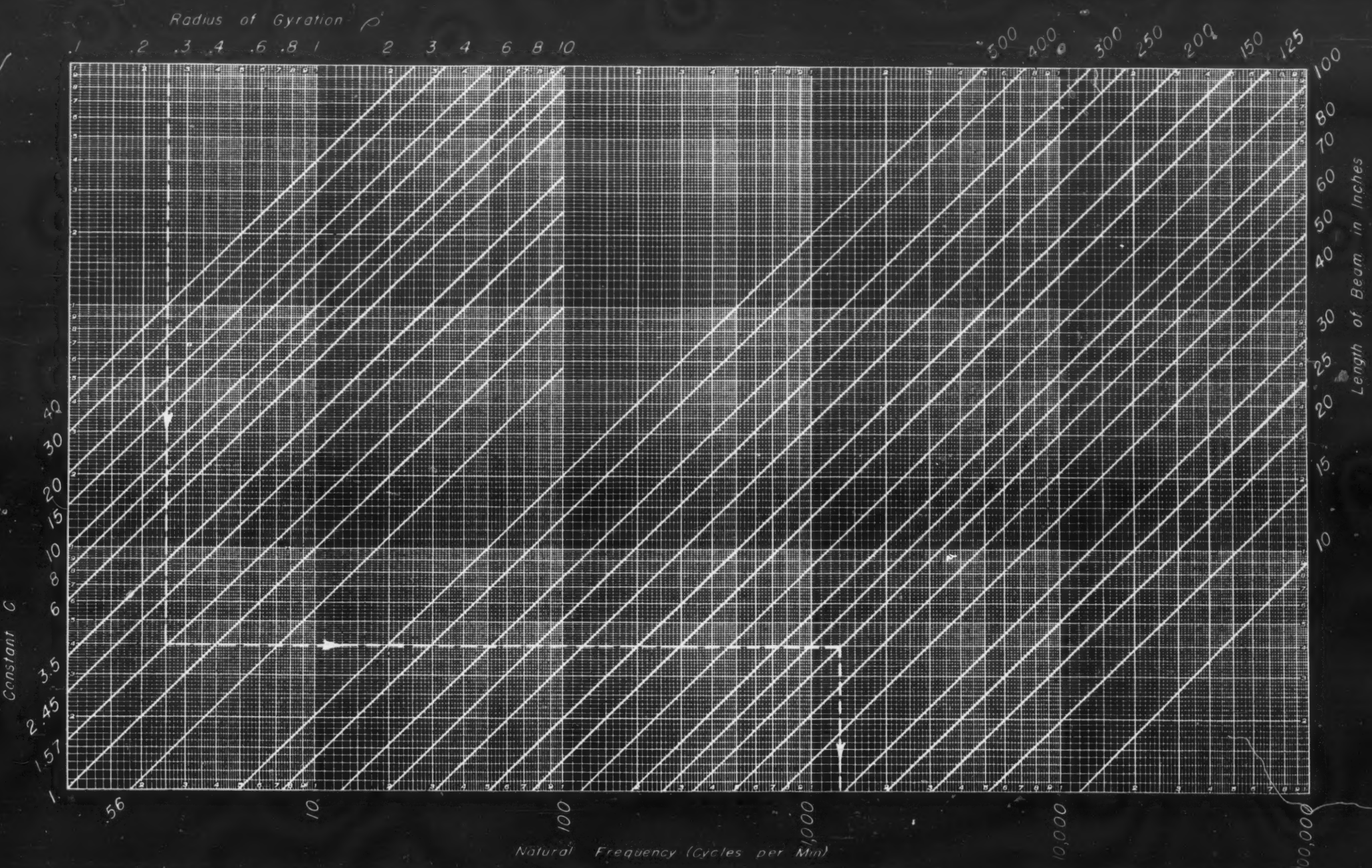


Fig. 1—Below—Modes of vibration up to the fifth

Fig. 2—Opposite Page—Chart enables rapid determi-

(*Concluded from Page* 159)

where f = Natural frequency in cycles per minute
ρ = Radius of gyration of section = $\sqrt{I/A}$
A = Cross-sectional area of section.

For an aluminum beam Equation 1 reduces to

$$f = 1{,}210{,}000C\frac{\rho}{L^2}$$

For normal calculations an average value

$$f = 1{,}200{,}000C\frac{\rho}{L^2}$$

can be used for either steel or aluminum beams so that only one chart is needed.

The radius of gyration can usually be obtained from handbooks or manufacturers' catalogs. TABLES I and II list typical values of radius of gyration for tubular members and rods frequently used for controls. The

TABLE I

Radius of Gyration for Round Aircraft Tubes

Diameter	Wall Thickness	ρ
⅜	.035	.121
½	.035	.165
⅝	.035	.209
¾	.035	.253
⅞	.035	.297
1	.035	.341
1⅛	.035	.385
1¼	.035	.429
¾	.049	.248
1	.049	.337
1¼	.049	.425
1½	.049	.513
1¾	.049	.602
2	.049	.690
2½	.049	.866

values for structural members and other sections can be obtained from the manufacturers' and handbook tables. For all members the radius of gyration can be calculated from the relation

$$\rho = \sqrt{\frac{I}{A}}$$

To give a rapid method for calculating the natural frequency of the various beams and modes for which the C values are given in *Fig.* 1, *Fig.* 2 has been constructed. The use of this chart can best be illustrated by working a typical example.

ENGINEERING DATA SHEET

EXAMPLE: Determine the natural frequency of a ¾-inch by .035-inch control tube 60 inches long and pin-connected at the ends. The value ρ is found from TABLE I to be .253 and the C value is found from *Fig.* 1 to be 1.57. The value of the natural frequency is found from *Fig.* 2 by entering the chart at the top at the value ρ = .253, tracing down to value of $C = 1.57$, then across to a length of 60 inches and down, and reading the natural frequency as 1320 cycles per minute on the lower scale. This example is illustrated in heavy lines on the figure. Higher natural frequencies are 5280, 11,860 and other higher values, corresponding to $C = 6.28$, 14.1, etc. If there are shaking forces of these frequencies serious vibration is likely to be encountered. This may in turn cause the member to fail by fatigue, particularly near the end fitting, or it may interfere with the accuracy of the adjustment if it is used in a control device.

TABLE II

Radius of Gyration for Rods and Shafts

Diameter	ρ
⅛	.062
[illegible]	[illegible]
⅜	[illegible]
[illegible]	.1[illegible]
½	.125
⅝	.156
¾	.1875
⅞	.219
1	.250
1¼	.3125
1½	.375
1¾	.4375
2	.500
2½	.625

REFERENCES

1. Lord Rayleigh—"Theory of Sound"—Macmillan and Co. Ltd., London, England.
2. C. R. Freberg and E. N. Kemler—"Aircraft Vibration and Flutter", John Wiley & Sons Inc., New York, 1944, from which *Fig.* 1 in this data sheet is reproduced by permission of the publishers.

MACHINE DESIGN'S

MATERIALS WORK SHEET

FILING NUMBER
12.01

Silicon Bronzes

ASTM Nos. B96-42, B97-41, B98-42 and B99-42

AVAILABLE IN:

(B96-42, Types A and C): Plate and sheet for pressure vessels
(B97-41, Types A, B and C): Sheet for general purposes
(B98-42, Types A, B, C and D): Rods, bars and shapes
(B99-42, Types A, B, C and D): General purpose wire

ANALYSES:

Type	Copper (min)	Silicon (range)	Lead (range)	Manganese* (max)	Zinc* (max)	Iron* (max)	Tin* (max)	Lead* (max)
A	94.8	2.75 to 3.5	.....	1.5	1.5	1.6	.75	.05
B	96.0	.75 to 2.0†	.....	.75	1.5	.8	1.6	.05
C	91.0	2.00 to 3.0	.....	1.25	4.5	.8	...	.05
D	94.0	2.75 to 3.5	.2 to .8	1.5	1.5	.25	...	...

*One or more of these elements may be present within the limits specified. Sum of all elements listed shall not be less than 99.5 per cent.

†Alloy having 2.6 per cent maximum of silicon is acceptable if sum of all elements other than copper and silicon and iron does not exceed .3 per cent.

Plate and sheet, covered by ASTM No. B96-42 and available in Types A and C, may, in addition to above analyses, contain .6 per cent maximum of nickel.

General purpose sheet, covered by ASTM No. B97-41, is available in Types A, B and C. Analyses are as above except that for Types A and C a maximum of .6 per cent nickel may be present.

PROPERTIES

TENSILE STRENGTH

(1000 *psi—range and minimum values*)

Temper	Form	Types A	B	C	D
Hot rolled	Plate and sheet for pressure vessels	55	..	55	..
Hot Rolled, cold-rolled finish	Plate and sheet for pressure vessels	58-72	..	58-72	..
Annealed	Plate and sheet for pressure vessels	50-67	..	50-67	..
	Wire, ¼-in. and under, diam	55-70	43-55	55-70	55-70
	Wire, over ¼-in. diam	55-65	43-55	55-65	55-65
Soft	All rods, bars and shapes	52	40	52	52
⅛ Hard	Wire, all diameters	62-78	50-65	62-78	62-78
.070-mm anneal	Sheet for general purposes	52-58	38-45	52-58	..
.040-mm anneal	Sheet for general purposes	55-64	40-50	55-64	..
¼ Hard	All rods and shapes	55	..	55	55
	Wire, all diameters	72-85	60-75	72-85	72-85
	Sheet for general purposes	62-72	42-52	62-72	..
½ Hard	Sheet for general purposes	71-81	..	71-81	..
	Plate and sheet for pressure vessels	115	90	115	115
	Rods up to 2-in. diam or across flats	70	55	70	70
	All Shapes	[1]	[1]	[1]	[1]
	Wire, all diameters	90	75	90	90
Hard	Sheet for general purposes	87-97	60-70	87-97	..
	Rods up to 1½-in. diam or across flats	85	65	85	85
	Rectangular bars up to 1-in. wide	65	[1]	65	65
	1 to 1½-in. wide	60	[1]	60	60
	1½ to 3-in. wide	50	[1]	50	50
	Wire, all diameters	115	90	115	115
Extra hard	Sheet for general purposes	99-108	67-76	99-108	..
	Rods up to ½-in. diam or across flats	100	85	100	100
	Rods ½ to 1½-in. diam or across flats	..	75	..	..
Spring	Sheet for general purposes	105-113	71-79	105-113	..
	Wire, ¼-in. and under, diam	135	100	135	135
	Wire, over ¼-in. diam	130	100	130	130

[1] Requirements of these forms are agreed upon by manufacturer and purchaser at time of order.

MACHINE DESIGN is pleased to acknowledge the collaboration of the following companies in this presentation: The American Brass Co.; Bridgeport Brass Co.; Chase Brass and Copper Co.; Revere Copper and Brass Inc.; Scovill Mfg. Co.

YIELD STRENGTH

(.5% strain, 1000 psi, min values)

Temper	Form	Types A	B	C
Hot rolled	Plate and sheet for pressure vessels	20	..	20
Hot rolled, cold-rolled finish	Plate and sheet for pressure vessels	29	..	29
Annealed	Plate and sheet for pressure vessels	18	..	18
Soft	All rods bars and shapes	15	12	15
¼ Hard	All rods and shapes	24	..	24
½ Hard	Plate and sheet for pressure vessels	40	..	40
	Rods up to 2-in. diam or across flats	38	20	38
	All shapes	[1]	[1]	[1]
Hard	Rods up to ½-in. diam or across flats	50	35	50
	All shapes	[1]	[1]	[1]
Extra hard	Rods up to ½-in. diam or across flats	55	55	55
	Rods ½ to 1-in. diam or across flats	..	45	..
	Rods 1 to 1½ in. diam or across flats	..	40	..

[1] Requirements of these forms are agreed upon by manufacturer and purchaser at time of purchase.

BENDING PROPERTIES

(angle of bend, degrees)[2]

Temper	Form	Types A	B	C
Soft	All rods and shapes	180	180	180
¼ Hard	All rods and shapes	120	....	120
½ Hard	All rods	120	180	120
	Shapes	[1]	[1]	[1]
Hard	All rods	90	120	90
	Shapes	[1]	[1]	[1]
Extra hard	All rods	60	90	60

[1] Requirements of these forms are agreed upon by manufacturer and purchaser at time of purchase.
[2] Angle through which a bend-test specimen can be bent cold to a radius equal to its diameter or thickness without fracturing.

ELONGATION IN 2 INCHES

(min, per cent)

Temper	Form	Types A	B	C
Hot rolled	Plate and sheet for pressure vessels	50	..	50
Hot rolled, cold-rolled finish	Plate and sheet for pressure vessels	20	..	20
Annealed	Plate and sheet for pressure vessels	40	..	40
	Wire, ¼-in. and under, diam	40	33	40
	Wire, over ¼-in. diam	47	40	47
Soft	All rods, bars and shapes	45	35	45
⅛ Hard	Wire, ¼-in. and under, diam	20	12	20
	Wire, over ¼-in. diam	28	20	28
¼ Hard	All rods and shapes	30	..	30
	Wire, ¼-in. and under, diam	15	8	15
	Wire, over ¼-in. diam	18	15	18
½ Hard	Plate and sheet for pressure vessels	10	..	10
	Rods up to ½-in. diam or across flats	20	11	20
	Rods ½ to 1-in. diam or across flats	25	14	25
	Rods 1 to 2-in. diam or across flats	30	17	30
	All shapes	[1]	[1]	[1]
	Wire, ¼-in. and under in diam	4	4	4
	Wire, over ¼-in. diam	10	10	10
Hard	Rods up to ½-in. diam or across flats	14	8	14
	Rods, ½ to 1-in. diam or across flats	18	10	18
	Rods, 1 to 1½-in. diam or across flats	22	12	22
	Bars up to 3-in. diam	25	..	25
	All shapes	[1]	..	[1]
	Wire, ¼-in. and under, diam	2	3	2
	Wire, over ¼-in. diam	6	8	6
Extra hard	Rods up to ½-in. diam or across flats	7	6	7
	Rods, ½ to 1-in. diam or across flats	..	8	..
	Rods, 1 to 1½ in. diam or across flats	..	10	..
Spring	Wire, ¼-in. and under, diam	1.5	2	1.5
	Wire, over ¼-in. diam	4	6	4

[1] Requirements of these forms are agreed upon by manufacturer and purchaser at time of order.

ROCKWELL HARDNESS

Temper	Form	Types A — Scales — F	A — B	B — F	B — B	C — F	C — B
Hot rolled	Plate and sheet for pressure vessels	72 min				72 min	
Hot rolled, cold-rolled finish	Plate and sheet for pressure vessels		60-80				60-80
Annealed	Plate and sheet for pressure vessels	72-92				72-92	
.070-mm anneal	Sheet for general purposes	70-82		45-55		70-82	
.040-mm anneal	Sheet for general purposes	76-93		50-75		76-93	
¼ Hard	Sheet for general purposes		65-80		48-63		65-80
½ Hard	Plate and sheet for pressure vessels		79-91				79-91
	Sheet for general purposes		79-91		64-73		79-91
Hard	Sheet for general purposes		88-96		74-82		88-96
Extra hard	Sheet for general purposes		93-98		78-85		93-98
Spring	Sheet for general purposes		94-99		81-86		94-99

PHYSICAL CONSTANTS

	Types A	B	C	D
Melting Point (deg C)	993-1027	1021-1060	993-1030	1019-1027
Specific Gravity	8.54-8.58	8.72-8.79	8.54-8.63	8.53-8.66
Weight (lb per cu in.)	.308-.310	.313-.317	.308-.310	.308-.313
Electrical Conductivity (% of I.A.C.S. at 20 C)	6 to 8.6	8.5 to 15	6 to 8.6	6.2 to 8.6
Ave. Coef. of Linear Expansion 68 to 572 F (per deg F)	.00001	.0000099	.00001	.00001
Thermal Conductivity (cal/cm²/cm/sec/deg C) at 20 C	.075-.095	.10-.215	.075-.115	.075-.095
(Btu/ft²/ft/hr/deg F) at 68 F	18-23	24-52	18-28	18-23
Modulus of Elasticity (psi)	15,000,000	17,000,000	15,000,000	15,000,000

Note: Constants are given in ranges to cover all commercial alloys produced within the analysis ranges of the four standard types.

CHARACTERISTICS

Silicon bronzes (copper-silicon alloys) usually contain one or more other elements as indicated on the first page of this work sheet under "Analyses." These alloys are nonmagnetic and highly resistant to fatigue. They possess the strength of mild steel and, in addition, offer excellent corrosion resistance. They can be stamped, spun, forged, pressed either hot or cold, and welded by almost all of the usual methods. They are work-hardening alloys. Thus, by cold working, their tensile strength and hardness can be brought to a maximum.

Of the four types, A and B see most general usage. Type A is used in those applications which require the best tensile properties in combination with resistance to corrosion equal to or better than copper. Welding characteristics of Type A are similar to those of mild steel. Type B alloys are characterized by exceptionally good cold-working properties, tensile properties comparable to 85-15 brass, corrosion resistance on a par with copper, and welding properties slightly inferior to Type A. Work-hardening occurs much more rapidly and results in better tensile properties for Type A than for Type B.

APPLICATIONS

Type A: Sheet and plate are used for fuel tanks, tumbling barrels, ducts, etc., where good strength plus excellent corrosion resistance is required. In the form of hard-rolled strip, it has good spring qualities. Rod and wire of this alloy find widespread use in bolts, nuts, washers, etc. Hot-rolled and annealed plates are particularly suitable for unfired pressure vessels.

Type B: Excellent material for machine screws, bolts, nuts, washers, gears and shafts for such items as washing machines, home garbage disposal equipment, laundry machines, water meters and water-pressure regulators. It is used also for needle valves and butterfly shafts for automobile carburetors.

Type C: Vessels for storage or processing of water, oil and various other liquids; for fans, blowers, smoke washers, ducts, humidifiers and agitators; also for baskets, screens, filters, pipe, springs, electrical tubing, etc.

Type D: Similar to Type A, primary difference being that it contains a small amount of lead which is added to improve machining characteristics. Rods and wire of this alloy are used extensively in the manufacture of screw-machine products requiring good strength and corrosion resistance.

FABRICATION

MACHINABILITY:

Types A, B and C, being strong and tough, have about 30 per cent of the machinability of free-cutting yellow brass. Chips break up with difficulty and form long spirals. Considerable tool pressure is required to separate the chip from the original stock and this generates

heat which is detrimental to tool life.

A tool suitable for machining mild steel generally is satisfactory if provided with liberal rake, clearance and lip angles. Best results are obtained when proper cutting compounds are used. For roughing operations a good grade of water soluble oil generally is used, while for finishing operations and drilling, tapping and milling, a sulphurized oil seems best. In any case the cutting compound is generously applied during machining, then thoroughly removed from the finished pieces. Lathe and boring practice, generally speaking, is about the same as for steel. Tools are rigidly supported and set not too much above center.

Type D silicon bronze, because it contains a small amount of lead, is more easily machined than Types A, B and C. It has a machinability rating of about 60 compared to 100 for free-cutting yellow brass.

DRILLING:

While short-spiral (about 40-degree helix angle) drills are recommended, standard drills often are used with undercut cutting edges. "Lands" of the drills should be narrow and polished. Continuous cutting is essential because if the drill is permitted to "dwell," the material at the bottom of the hole will glaze and work-harden, not only making it difficult to resume cutting but dulling the drill as well. Ample cutting oil should be applied. Where large, deep holes are to be drilled, the use of drills having integral oil holes is recommended.

TAPPING:

For general work, a tap-drill size which will result in about a 75 per cent depth of thread should be specified. Attempts to tap full-depth threads in Types A, B and C silicon bronze often result in torn threads, unnecessarily high tap breakage and the use of excessive power. In addition, there is but slight difference between the strength of a 75 per cent thread and a full-depth thread. Taps should have narrow lands and deep, polished flutes with rake angles of from 5 to 8 degrees. End chamfer is best made relatively short (about two threads) to obviate work-hardening of the material. Tapping speeds—using high-speed steel taps—range from 20 feet per minute for coarse threads to 60 feet per minute for fine threads. Copious supply of sulphurized oil is recommended for all tapping operations.

REAMING:

In all cases and particularly where wall thickness is small in relation to hole diameter, size of the drilled hole preceding reaming should leave sufficient stock to permit the reamer a good "bite." Where insufficient stock is allowed, the reamer burnishes the material, resulting in an undersized hole and excessively rapid wear of the reamer. Lands should be narrow and flutes well polished to assure smooth finishing. Standard spiral-fluted reamers with a helix angle of 7 to 12 degrees work well, and left-hand spiral, right-hand cutting reamers give excellent results. Where high-speed steel reamers are employed, the cutting speed ranges from 75 to 150 feet per minute. As with drilling and tapping, ample cutting lubricant is used.

STAMPING AND PERFORATING:

With the exception of Type D, these alloys are used for a wide variety of stamping and perforating operations. In general they work much like various brass alloys for such operations. Because of the wide variation in design of parts, users should furnish the metal manufacturer details of the size and shape of the part to be fabricated, so that material of the correct temper can be supplied.

DEEP DRAWING:

Copper-silicon alloys in types A, B and C are rated as having excellent cold-working properties. In such a specialized operation as deep drawing, however, the utmost in working qualities usually is required. Although these alloys are satisfactory for deep-drawing operations, they are not considered quite as satisfactory for the severest operations as the usual grades of yellow brass. Type B alloys can be drawn with fewer anneals during a sequence of operations, but their strength is not as high as the other types. Metal required for deep drawing should be specified as such in order that material of a correct degree of temper can be supplied.

SPINNING:

Copper-silicon alloys in types A, B and C are used for spinning operations, but here again it is essential that metal for such purposes be specified as such, in order that the material can be furnished in the proper annealed temper. In general, these alloys are rated as having good spinning qualities but cannot be spun with quite the facility of the usual brasses employed for such work. In this type of cold-working operation, the amount of reduction possible without annealing will vary considerably with the alloy, the shape of the part, and the size and thickness of the blank.

FORGING:

With the exception of type D, these alloys are suitable for hot working and are rated excellent in this respect. The preferred temperature range for hot forging is 1200 to 1500 degrees Fahr. Types A and C usually are used, since type B has lower strength in the hot-forged condition and is furnished as being especially suited to cold-forging operations such as cold-heading. These alloys can be furnished also in the form of hot-pressed forgings made in closed dies, providing they are of rather simple design; the alloys do not flow into intricate dies as readily as other types of copper alloys used for hot-pressed forgings of complicated design.

WELDING:

Silicon bronzes, except Type D, can be welded by almost all standard methods and can be brazed also. Same material as the base metal is used for both filler rod and electrodes. Preparatory to welding, all scale, dirt, oil, grease, tarnish and oxide is removed by filing, scraping or grinding. A good flux is 90 per cent fused borax and 10 per cent sodium fluoride finely ground, either in powder form or a paste made by mixing the powder with alcohol. For butt welding sheets 3/16-inch and thinner, no beveling is necessary. Beveling is necessary, how-

ever, for thicker sheets.

Carbon-arc welding is most widely used because it is rapid, inexpensive and highly satisfactory. Further, it is silent, does not spatter, and in general is more satisfactory than the metallic arc. For flanged edges of thin metal the edges can be fluxed and melted down without using filler rod. Arc length is held to about ½-inch and care is exercised not to overheat.

When oxyacetylene welding is employed, a slightly oxidizing soft flame is prescribed and both joint and filler rod are fluxed. Seam and spot welding are highly satisfactory for making structural shapes such as angles, channels, trusses, tanks, etc. Lighter pressures, shorter time and slightly greater current are required than for welding steel. Metallic-arc welding is satisfactory for thin gages. Coated filler rod with reverse polarity (metallic electrode positive) is used. Weld is fluxed lightly.

RESISTANCE TO CORROSION

Like copper, these alloys can be used satisfactorily in contact with a number of corrosive solutions and compounds. It is well to bear in mind, however, that this corrosion resistance may be adversely affected by oxidizing agents. Generally speaking, these alloys may be used in practically all locations where copper is suitable but where greater strength than copper is required. They offer good resistance to atmospheric corrosion in both industrial and marine locations, to corrosion by sea water as well as corrosive industrial and natural waters, to sulphuric and hydrochloric acid and many of their salts, to alkalis and many of the alkali salts. They show good resistance also to certain organic compounds including alcohol, phenol, cresol, sugar solutions, fatty acids, and organic salts.

GALVANIC CORROSION

These alloys will exhibit galvanic corrosion effects similar to pure copper. In general, they may be safely coupled with other copper alloys. Where large areas are connected galvanically to iron, zinc, aluminum, or high-brass in a corrosive environment, the whole assembly should be protected with paint or other insulation.

ANNEALING

Types A, B and D alloys will recrystallize after severe cold reduction at approximately 750 degrees Fahr. and at somewhat higher temperatures following more moderate reductions. The annealing range suggested is from 900 degrees Fahr. for light anneals to 1300 degrees Fahr. for full anneals, depending somewhat on the size or thickness and the degree of softness desired. The oxide formed during annealing is particularly tenacious and the usual brass mill pickle solution containing 10 to 20 per cent sulphuric acid leaves these alloys with a thin oxide coating of dark red color.

MATERIAL DESIGNATIONS

ASTM No.	ASME Boiler Code	Federal	U. S. Navy
B96-42, Type A	SB-96, Type A	....	....
Type C	SB-96, Type C	....	....
B97-41, Type A	....	QQ-C-591a, Class A	46B27 (INT) Oct. 15, 1943
Type B	....	....	....
Type C	....	....	....
B98-42, Type A	SB-98, Type A	QQ-C-591a, Class A	46B27 (INT) Oct. 15, 1943
Type B	SB-98, Type B	QQ-C-591a, Class B	....
Type C	SB-98, Type C	....	....
Type D	SB-98, Type D	QQ-C-591a, Class D	46B27 (INT) Oct. 15, 1943
B99-42, Type A	....	....	....
Type B	....	....	....
Type C	....	....	....
Type D	....	....	....

MATERIAL TRADENAMES

ASTM No. B96-42

Types A	B	C	D	Producers
Everdur 1010	....	....	....	The American Brass Co.
Duronze II	....	....	....	Bridgeport Brass Co.
Olympic Bronze Type A	....	....	....	Chase Brass and Copper Co.
PMG10	....	PMG10, 94, 96	....	Phelps Dodge Copper Prod. Corp.
Herculoy 418	....	....	....	Revere Copper and Brass Inc.
Cusiloy	....	Cusiloy	....	Scovill Manufacturing Co.

ASTM No. B97-41

Everdur 1010	Everdur 1015	....	....	The American Brass Co.
Duronze II	Duronze I and V	....	....	Bridgeport Brass Co.
Olympic Bronze Type A	....	....	....	Chase Brass and Copper Co.
PMG10	PMG3	PMG10, 94, 96	....	Phelps Dodge Copper Prod. Corp.
Herculoy 418	Herculoy 421	....	....	Revere Copper and Brass Inc.
Cusiloy	....	Cusilov	....	Scovill Manufacturing Co.

ASTM No. B98-42

Everdur 1010	Everdur 1015	....	Everdur 1012	The American Brass Co.
Duronze II	Duronze I and V	....	....	Bridgeport Brass Co.
Olympic Bronze Type A and W	Olympic Bronze Type B	....	....	Chase Brass and Copper Co.
PMG10	PMG3	PMG10, 94, 96	PMG95	Phelps Dodge Copper Prod. Corp.
Herculoy 418	Herculoy 421	....	....	Revere Copper and Brass Inc
Cusiloy	....	Cusiloy	....	Scovill Manufacturing Co.

ASTM No. B99-42

Everdur 1010	Everdur 1015	....	Everdur 1012	The American Brass Co.
Duronze II	Duronze I and V	....	....	Bridgeport Brass Co.
Olympic Bronze Types A and W	Olympic Bronze Type B	....	....	Chase Brass and Copper Co.
PMG10	PMG3	PMG10, 94, 96	PMG95	Phelps Dodge Copper Prod. Corp.
Cusiloy	....	Cusiloy	....	Scovill Manufacturing Co.

PROFESSIONAL VIEWPOINTS

". . . key to bearing life"

To the Editor:

In his article "Surface Finish—Key to Bearing Life", published in the November issue of MACHINE DESIGN, Mr. Hemingway made a valuable contribution to bearing design. Accurate test data on bearing performance are needed and those which eliminate all variables except surface condition of the shaft are particularly valuable because they fill a gap in our knowledge of bearing performance.

The author's illustrations indicate that the bearing operated under fluid friction condition up to a certain load because the bearing temperature remained constant. At this certain load point, which differed for different surface conditions of the shaft, fluid friction ceased and metal-to-metal contact took place. On account of the sudden temperature rise at this point, further operation of the bearing was impossible. These tests are therefore another proof of the approximate validity of the Petroff equation for bearing operation, as expressed by the formulas

$$H = \frac{D^3 L n^2 Z}{1.04(10)^3 C}$$

$$f = \frac{DZn}{2(10)^7 Cp}$$

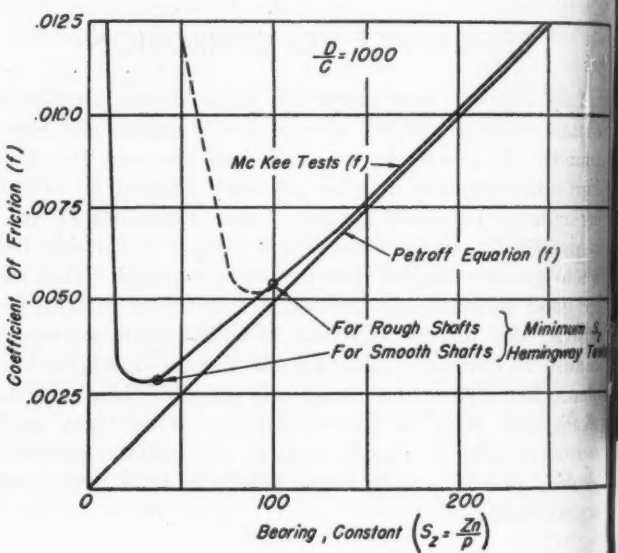
where H = heat developed in bearing, Btu/hr, f = coefficient of friction, D = bearing diameter in inches, L = bearing length in inches, Z = oil viscosity in centipoises, n = rotational speed, revolutions per minute, C = diametral clearance in inches, P = projected bearing pressure, pounds per square inch, and $Zn/p = S_z$ = a constant that is dimensionless.

Fluid Friction Operation

The tests prove that the friction H produced by the bearing is virtually independent of the load under fluid-friction condition. They also prove that the surface roughness of the shaft affects the range of fluid-friction operation considerably. For shafts with surface roughness of 35 microinches the minimum figure for the bearing constant seems to be S_z = 100. For shafts ground to from 2 to 12 microinches the minimum figure is apparently S_z = 15 to 30.

From the graphical illustration of the Petroff equation it may be deduced that smooth shafts allow higher unit loadings and smaller bearing constants and, provided the proper constant S_z is selected, also give lower friction figures. In view of the fact that heat dissipation is the most critical item in many bearing problems, this knowledge is extremely important.

The tests also indicate that it may be unnecessary to go to too much refinement in finishing the shaft and sleeve surface of bearings operating under fluid-friction condi-



From "Lubrication", by A. E. Norton

Graphical illustration of the apparent proximity of the theoretical Petroff equation and test results obtained by McKee and Hemingway

tion. The shaft floats on an oil film and does not, except at starting, have metal-to-metal contact. The operation follows Newton's law of viscous flow in which surface roughness is of no importance. Above the minimum figure for the bearing constant S_z, the surface condition of the shaft or sleeve is therefore without influence. Both should be reasonably smooth, however, to give low friction figures for starting.

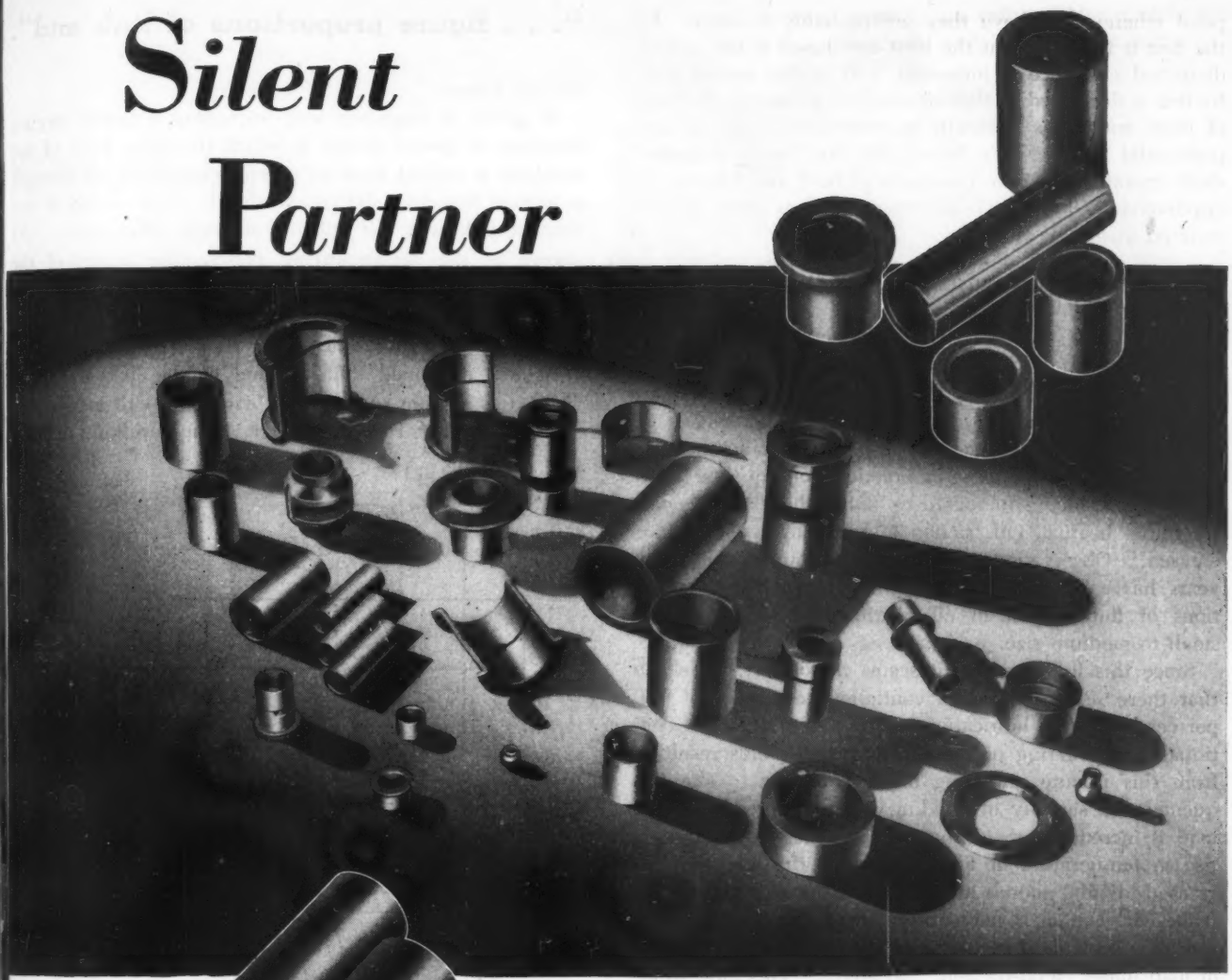
—H. W. HAMM,
York, Pa.

To the Editor:

I have read Mr. Hamm's comments on my article, "Surface Finish—Key to Bearing Life", with more than usual interest. They show that he has made a thorough analysis of the graphs contained therein and I believe that his conclusions are essentially correct. I would like to qualify somewhat, the comments he has made.

With reference to use of Petroff's formulas, I note that

Silent Partner



of Performance

QUIET operation is a strong selling point for all types of household appliances. One easy way to eliminate noise is to install Johnson Bronze Sleeve Bearings in the motive unit. They are precision made products, guaranteed to deliver the utmost in performance for the longest period of time.

Manufacturers giving thought to their postwar product will do well to consult with Johnson Bronze now. Simply call in a Johnson Engineer. Permit him to review your applications. All of his recommendations will be based on facts, supported by more than thirty-five years exclusive bearing experience. There is a Johnson Engineer—as near as your phone.

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BRANCHES IN 18 INDUSTRIAL CENTERS

point wherein I believe they are probably in error. In the first it is stated that the heat developed is less as the diametral clearance is increased, and in the second that friction is decreased as that clearance is increased. Neither of these conditions is strictly in accordance with our experimental results. We found that our 1-inch diameter shaft operated with the minimum of heat and friction at approximately .0007-inch clearance. Either more or less resulted in loss of efficiency.

Shaft Flutters in Bearing

In the writer's article it was mentioned that intermittent metallic contact was found in every test, even under the lightest loads. Evidently the shaft seldom rotated about its true axis but fluttered around within the bushing. Naturally this action is reduced by lowered clearance but becomes severe enough to be of practical consequence when the test clearance is increased to .002-inch. At that point we found heating and friction to be quite noticeably increased. Observation of machine bearings over many years has caused the writer to believe the same conditions of flutter exist in all bearings, at least those of small to medium size.

Since this flutter exists, it seems reasonable to believe that there seldom occurs a continuous condition of 100 per cent fluid-film lubrication. There is, therefore, a continual mild wearing of the shaft by tiny welds resulting from this minute pounding of the bushing. The frequency and severity of welding contact increases as the load is increased for, as shown by the tests, there was a rise in temperature of fifty degrees as the load was increased to 1500 pounds after the one hour of run-in. Thus, while Mr. Hamm is not entirely correct, for all practical purposes he is right in stating that heating is independent of load under "fluid-friction" conditions. This mild welding action actually increases shaft smoothness by slowly wearing away ridges and high spots.

Finish Affects Heating

The tests warrant the conclusion that excessive refinement of surface is unnecessary under fluid-film lubrication. However, it must be noted that both clearance dimension and seizure load capacity are limited by surface finish conditions. Furthermore, low starting friction and surface damage during starting and stopping require smooth surfaces. Heating increases rapidly as roughness becomes greater than about ten microinches. Mr. Hamm comments on the necessity for reduction of heat, and not the least damaging of the effects of heat is the distortion of the bearing surface shape.

Since the publication of the subject article in MACHINE DESIGN, the writer has done further work on the load-carrying capacity of soft shafts. While it is still believed that some depth of scratches is desirable for rendering a soft shaft more safely broken-in with the bushing, it has been found that even a soft shaft is not likely to be too smooth when alignment and conformity are exceptionally good. Indeed, a soft shaft Superfinished to two to three microinches shows about the same 60 per cent improvement in ultimate load-carrying capacity as a hardened shaft ground to ten microinches.

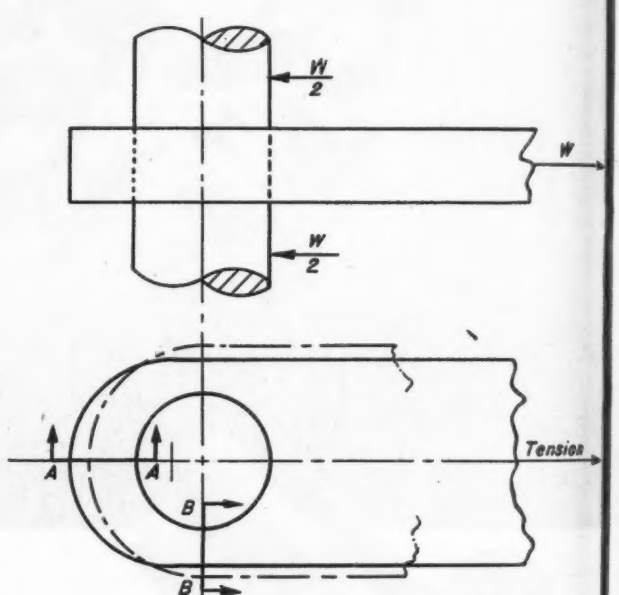
—E. L. HEMINGWAY, *Chief Metallurgist*
International Detrola Corp.

". . . figure proportions of link end"

To the Editor:

A group of engineers was inspecting a heavy forging machine of special design in which the main force of the machine is carried from an overhanging crankpin through a tension link to a sliding crosshead. The design of the tension link was the subject of some discussion. As shown in the accompanying sketch, the center of the radius of the end of the link was offset from the center of the pin, resulting in a larger area of metal through section *A-A* than through section *B-B*.

It was explained that the end of the link should be figured as a beam supported at each end with most of the load concentrated at the center. The resultant bending



Tension link detail showing alternate methods of designing stressed sections

stresses would justify the heavy section at *A-A*. A textbook formula supported this view by giving proportions for offset end radius as shown.

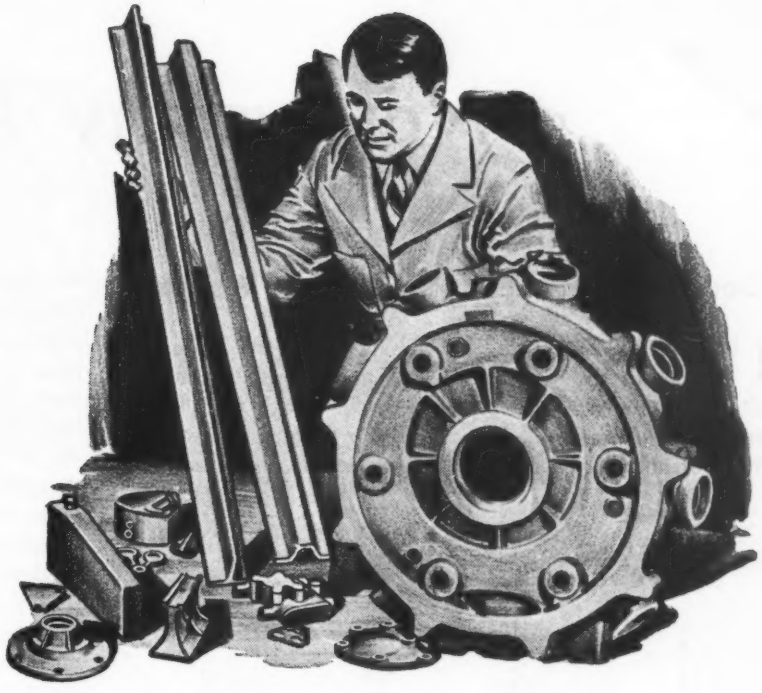
The opposite view was that when the pin is a reasonably close running fit in the link, the deflection in the "beam", and hence the bending stress, is negligible. Also, that such bending stresses as do occur would be felt at section *A-A* and combine with the tensile stress. The latter opinion is supported by maintenance experience on similar link and pin applications. Even when the link radius is concentric with the pin center as shown by dot-dash lines on the sketch, breakage usually occurs close to section *B-B*.

Authoritative suggestions from your readers on how to figure the proportions of this type of link end would be appreciated.

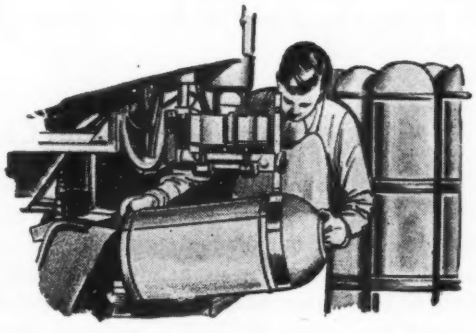
—E. S. H.

Readers of MACHINE DESIGN are invited to contribute solutions to this problem. It may be pointed out that the stress distribution is affected by deformation of the part, hence solutions should take this fact into account.—ED.

Magnesium alloys are fabricated and joined by all common methods



and Dow's complete fabrication plants are always ready to work with you



Dowmetal Magnesium brings you its incomparable lightness—without equal among structural metals—in the full range of common fabrication methods.

The fruits of more than a quarter-century of Dow engineering and fabrication experience are ready at hand to translate your ideas into new, weight-saving products of Dowmetal—*the metal of motion*. For in Dow's own shops—and in numerous other plants as well—Dowmetal Magnesium Alloys, with their high strength-weight ratio, show full versatility in sand, permanent mold and die casting; sheet, plate and strip rolling; rod, bar, tube and special shape extrusion; press and hammer forging—and bending, drawing, pressing, spinning, blanking, punching and shearing.

Likewise Dowmetal is readily joined by all usual procedures—riveting and arc, spot, gas and flash welding.

There is no mystery to magnesium processing techniques: These lightweight alloys call for procedures very similar to those employed with other metals.

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MAGNESIUM DIVISION, THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN

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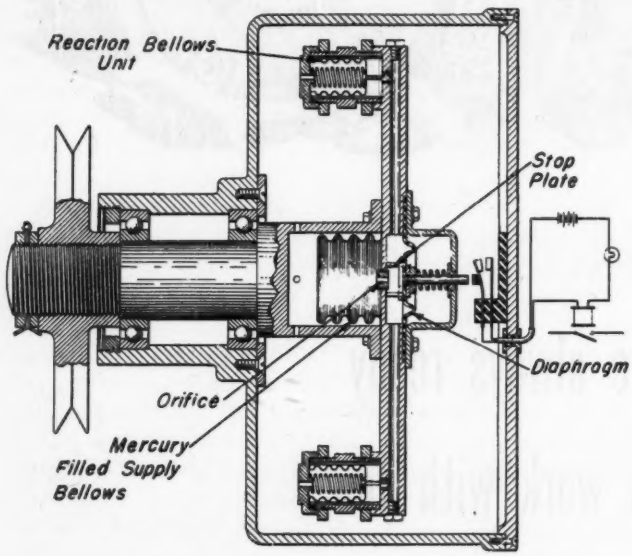
Noteworthy Patents

Decelerometer Controls Skidding Wheels

IT IS WELL known that the brakes of any vehicle will decelerate the wheel speed at an abnormally rapid rate to a locked condition when applied with sufficient force to overcome the adhesion or rolling friction between the tires and the road surface. This is undesirable since



Centrifugal force and reaction units in this mechanism are balanced against a specific orifice flow rate to detect excessive deceleration in vehicle wheels

less braking effort is obtained with sliding than with rolling friction conditions. Wheel maintenance under such conditions is high, especially in the case of railway wheels. The hydraulic decelerometer shown in the accompanying illustration was designed to overcome this braking problem and is covered by patent 2,347,962 recently assigned to the Westinghouse Air Brake Co. Mercury is utilized to provide a high density hydraulic medium for efficient operation. Materials employed in the system are necessarily those resistant to deterioration by the mercury.

The decelerometer unit as shown in the accompanying illustration is coupled to and driven by the wheel to be controlled during the braking cycle. Centrifugal force at normal operating speed draws the mercury out through the passages in the arms, compressing the reaction bellows units accordingly. Partial vacuum thus created at the orifice is overcome by the atmospheric pressure acting on the supply bellows. Mercury then passes through the orifice into the arms until the pressures are equalized. Collapse of the diaphragm under vacuum is prevented by the stop plate.

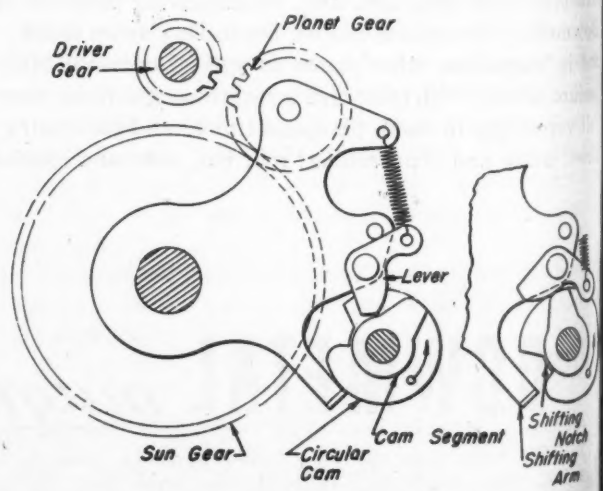
Braking of the driving wheels results in deceleration of the unit, reducing the centrifugal force in proportion to the square of the angular velocity of the driven shaft of the decelerometer. This creates a pressure differential between the mercury and the reaction bellows. Expansion of the spring in the reaction bellows forces a return of mercury to the supply unit. If the rate of return does not exceed the selected orifice flow rate the spring-loaded diaphragm remains unmoved. However, should the return rate of the mercury be substantially greater than the orifice flow rate the spring-retained diaphragm flexes to close the switch to an indicating lamp. Contact is maintained as long as the deceleration rate produces the predetermined pressure differential across the orifice.

Any relay controlled by the unit may be adapted to operate brake control equipment designed to maintain maximum deceleration without sliding. Control equipment on such vehicles as railway cars can thus retain, by intermittent braking, sufficient rolling speed to avoid flattened wheels produced by sliding stops. Although the unit is not sensitive to uniform deceleration, it detects different rates of deceleration at varying speeds to control slipping or sliding conditions common to many vehicles.

Gears Engage Without Clash

IN CERTAIN mechanisms it is highly desirable to utilize geared power transmission which can readily be shifted into and out of mesh without shock or clash. A design which provides resilient engagement of the gear teeth and also positive locking in the meshed position is covered by patent 2,339,154 assigned recently to the Monroe Calculating Machine Co.

Normally the planet gear of this arrangement is held disengaged from the driver by a shifting arm which rides upon a circular cam as shown in the accompanying illustration. Counterclockwise movement of the cam arrange-



Cam-operated mechanism utilizes a tension spring to produce resilient nonclashing engagement of gears

Lengthen Service Life
...regardless of unit size!

4 TIMES LONGER LIFE

Eight years' service instead of two, was obtained by a large chemical company through specifying Nickel cast iron in this 8-ton ring. Plain iron used here corroded and eroded ½" per year. This Nickel alloyed iron wears less than ⅛" in the same time.

Increases Result/Dollar Ratio...

In products and applications ... from a wrench to a railroad ... records show that a little Nickel goes a long way to lengthen product life, to reduce bulk and deadweight, and to save maintenance costs.

When compared with costs involved, metals to which Nickel is added invariably rise sharply in service value ... due to improved performance.

Nickel imparts strength, hardness, toughness, resistance to corrosion, heat and abrasive wear. In all fields of industry Nickel offers significant advantages. Send us details of your problem for our recommendations.

THE INTERNATIONAL NICKEL COMPANY, INC.
67 Wall Street, New York 5, N. Y.

☆ CONSERVE WITH NICKEL ☆

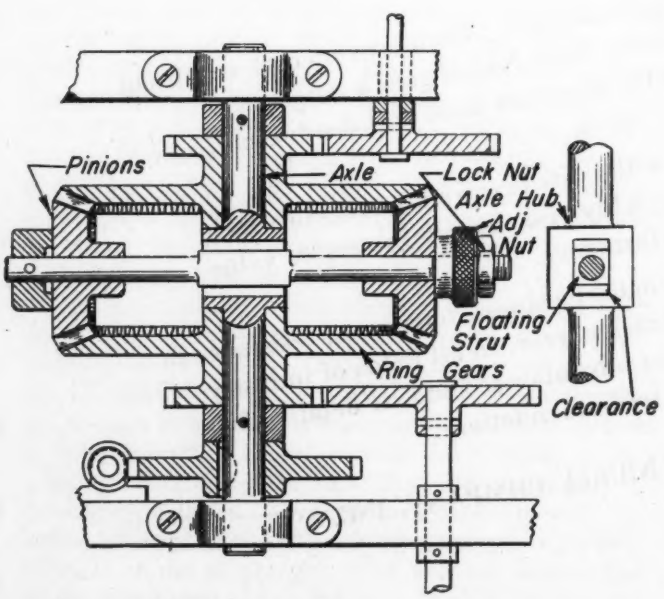
ment loads the tension spring by depressing the attached lever. Depression of this lever by the cam segment is completed as the shifting notch assumes the position shown in the detail at the right of the illustration. This frees the shifting arm from the cam and the planet gear is moved into meshing position by the reaction of the spring-loaded lever against the cam segment. Until complete engagement is established between the planet and driver gears, the spring yields in its action on the lever to allow meshing without clash. As the gears fully mesh, the lever pivots against the stop pin which aligns the flat end surfaces of both the lever and cam segment to retain the engagement during the power cycle. Further movement of the cam displaces the locking lever and returns the mechanism to the normal disengaged position.

Differential Is Self-Aligning

A SELF-ALIGNING pinion or planet gear assembly that will adapt itself for adjustment and the elimination of backlash or end play in differential gear assemblies is covered by patent 2,355,144 recently assigned to Jack & Heintz Inc.

With the construction shown in the accompanying illustration the pinions are self-centering and self-aligning with respect to the ring gears. Undue end play is removed by properly engaging the gear teeth by means of the adjustment nut provided. The floating strut acting as a shaft for the pinion gears provides vertical and pivotal adjustment in a single vertical plane that coincides with the axis of rotation of the axle. Thus, while the pinion gears are self-aligning and self-centering with respect to the ring gears, the tooth engagement is at all times tangential so that as end play is eliminated so also is the accompanying backlash.

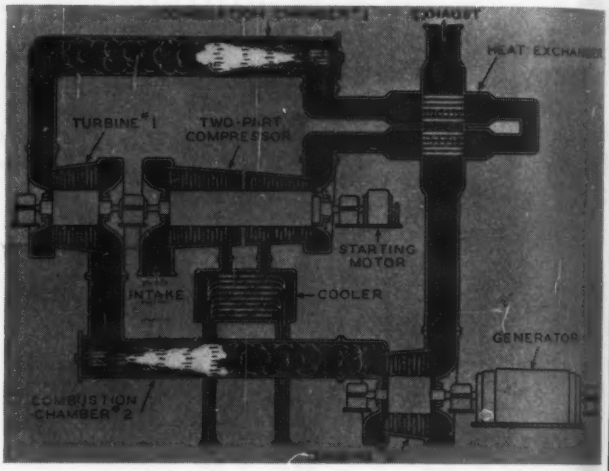
Importance of the accurate adjustment provided to eliminate control errors in the differential gearing is self-evident, especially in such precision instruments as this automatic pilot aileron and elevator control assembly.



Pinions operating on floating strut member are free to center and align properly with each ring gear, allowing for complete removal of backlash

Efficiency of Gas Turbines Must Be Increased!

ONE OF the most recent developments in prime power sources, the gas turbine, has gained wide acclaim and much popular appeal. However, it has been frankly indicated by such builders as Allis-Chalmers Mfg. Co. that the potentialities of the new prime



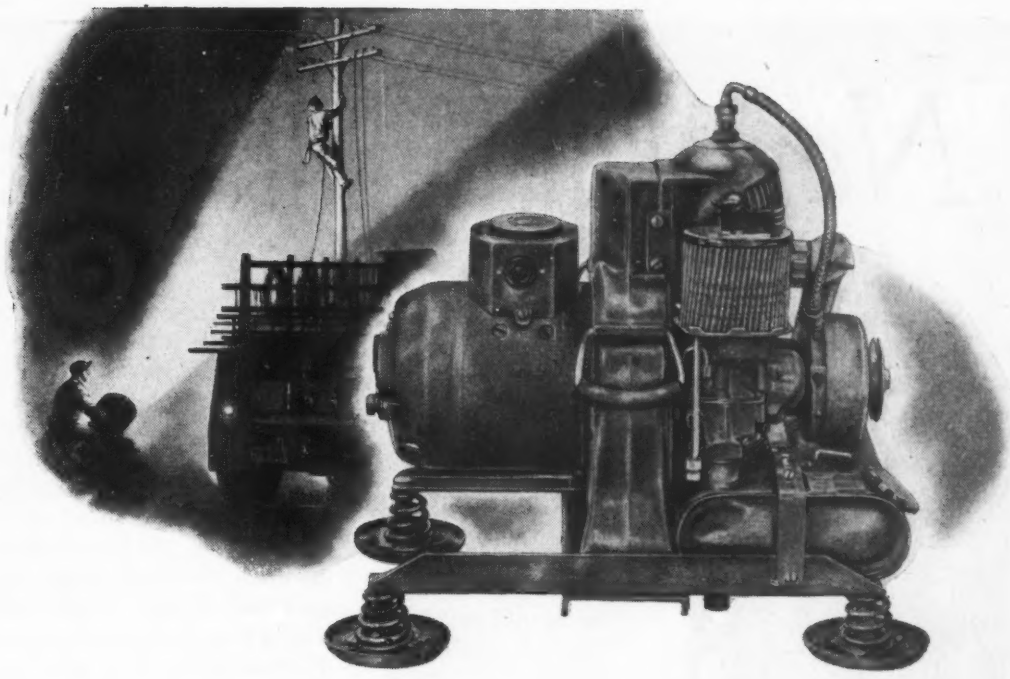
mover have not yet been completely realized. A great deal more of the extensive engineering effort already applied must still be put into development.

A representative unit, for example, produced a total output of 6500 horsepower. Five thousand horsepower of this output was consumed in the operation of the compressor. The balance, 1500 horsepower, was available for operation of a generator. Consequently, only 23 per cent of the total developed horsepower was available to be put to work, the remaining 77 per cent being consumed in maintaining the operating cycle. Increased efficiency within the unit thus appears to be the answer.

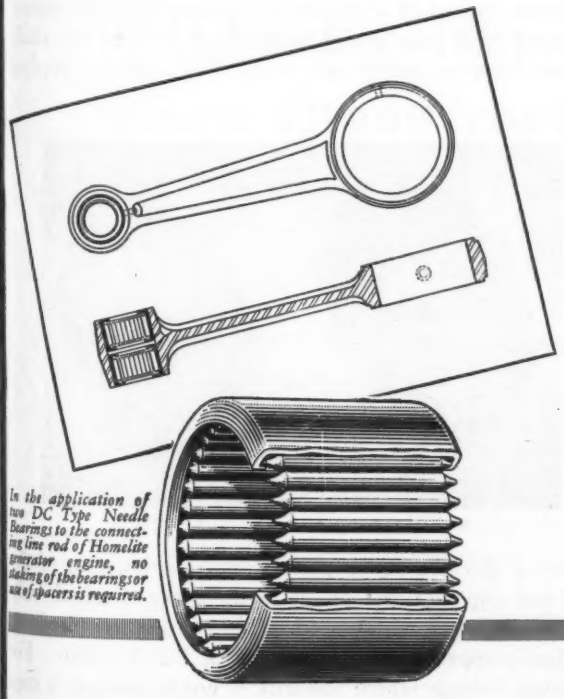
Some engineering expedients which have been evolved for increasing the efficiency of a gas turbine are shown in the accompanying illustration. The heat exchanger increases the thermal efficiency by utilizing the high temperature of the exhaust gases. Air, drawn through the exchanger, enters the combustion chamber at 475 degrees rather than the former 375 degrees without an exchanger. Thus, waste heat is partially reclaimed in the cycle and less fuel is required to heat the air to the necessary operating temperature of 1000 degrees.

Use of a second combustion chamber and turbine is another way found to increase the efficiency. The high-pressure turbine (No. 1) supplies power to the compressor only. Exhaust from the first turbine is reheated in the second combustion chamber to proper inlet temperature and then used to operate the low-pressure turbine. The entire output of the second turbine is then available for work to be done.

Reduction in the horsepower needed to operate the compressor affords the third approach to increased efficiency. A regenerative reheat cycle using a two-stage compressor with an intercooler is capable of increasing the thermal efficiency to 30 per cent in the compression cycle with turbine inlet temperature of 1200 degrees.



Torrington Needle Bearings Double Life on Wrist Pins of Homelite Portable Generators



In the application of two DC Type Needle Bearings to the connecting line rod of Homelite generator engine, no staking of the bearings or use of spacers is required.

The wrist pin bearing of a gasoline engine always takes a beating! To give maintenance-free operation under the impact and stress of extreme service conditions, it must have ample capacity, adequate lubrication, and be designed for long life.

Two DC Type Torrington Needle Bearings in the wrist pin end of the connecting rod of Homelite Portable Generators were found to give *double* the service life in this application.

Can the use of these compact, low-cost, anti-friction bearings increase the performance, improve the efficiency or lengthen the service-life of your equipment? Our engineering staff will be glad to review the matter with you and, if indicated, to work with you in selecting the size and type to meet your design requirements. Write us today. Or, if you wish more preliminary information, consult the Torrington Needle Bearing Catalog No. 30-A. We will gladly send you a copy if you do not already have one on hand.

THE TORRINGTON COMPANY

Established 1866

TORRINGTON, CONN. • SOUTH BEND 21, IND.

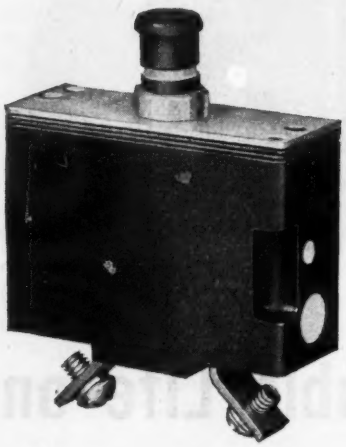
"Makers of Needle Bearings and Needle Bearing Rollers"

New York Boston Philadelphia Detroit Cleveland Seattle
Chicago San Francisco Los Angeles Toronto London, England

TORRINGTON NEEDLE BEARINGS

New PARTS AND MATERIALS

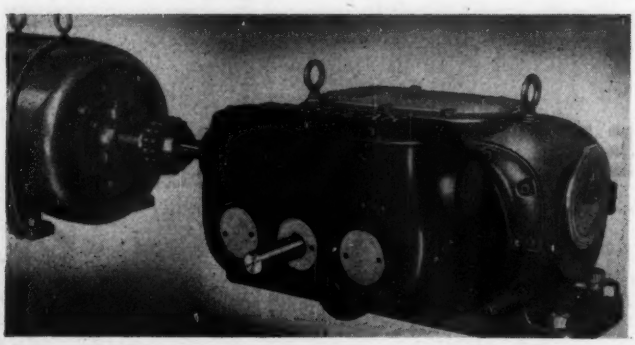
Push-Pull Circuit Breakers



ADDED TO THE LINE of toggle and push-type aircraft circuit breakers produced by Square D Co. Regulator Division, 6060 Rivard street, Detroit 11, the new Class 9311 Type A push-pull circuit breaker provides both switching and protection, and guards against accidental or careless operation of vital circuits. A definite pull action is required to open the breaker manually, providing a means of operation for servicing of aircraft on the ground. The manual reset button, black at top, is provided with a red hand to indicate trip in daylight and a luminescent band in darkness. A bimetallic trip element gives time limit protection against small momentary overloads, as well as instantaneous protection on heavy overloads and short circuit. After tripping, the reset button is pushed down to its farthest position for resetting. Although these breakers were especially designed for aircraft use, they are also suitable for boats, tanks and other mobile units having power sources up to 30 volts direct current.

Differential Variable-Speed Drive

CONSISTING OF two mechanical differentials and a standard variable-speed reducer, the new differential variable-speed drive of Ohio Gear Co., 1338 East 179th street, Cleveland, is available in many ranges for specific applications. It uses an 1800 revolutions-per-minute, squirrel-cage motor as a source of power, with an output speed from 0 to 800 revolutions per minute in one direction, or from 400 forward through 0 to 400 revolutions per minute in reverse. The distinctive feature of the drive is the fact that torque remains constant throughout the entire range. Torque capacity of the unit illustrated (No. T3) is 750 inch-pounds, and other units may be supplied to provide up to 2100 inch-pounds torque through the same speed ranges. Smaller units as well are available. By means of a control box furnished with the drive, the output shaft can be made to duplicate any movement of the control shaft, which may be operated with a few inch-pounds torque. In the two mechanical differentials the bevel gears are alloy steel, casehardened. Worms are hardened and ground, and the worm gears are gear bronze.



Drag Cup Induction Generators

NEW DRAG CUP induction generators have recently been made available by Electric Indicator Co., Stamford, Conn., in both base and frame-mounted models. The generators consist of a laminator stator-wound two-phase, stationary steel pole, and an aluminum cup on the shaft rotating between stator and pole. With voltage applied

to one of the two stator phase terminals, rotation of the shaft and cup induces voltage at the other terminal, voltage being lineal with speed. The torque required for rotation is approximately .25 grains at 1-inch radius. For increased voltage where linearity is not important, a copper cup may be used. Type 68 has an applied voltage of 115 volts at 60 cycles, alternating current, and generated voltage with resistive load 100,000 ohms varies from .15 volt maximum with cup stationary, to 1.2 volts minimum at 1000 revolutions per minute, with increase at uniform rate up to 6000 revolutions per minute. Type 101 generates from .15-volt maximum with cup stationary to 3.6 volts minimum at 980 revolutions per minute and 20

New leather packings, made a new way

VIM resin-impregnated "V" packings have these merits:

They are molded to shape, so that the grain, or smoother side of the leather, will contact the cylinder or plunger. This reduces friction, as compared to a trimmed O.D. and I.D. packing whereby the flesh side contacts the operating surface.

But that is only one advantage of VIM Resin-Impregnated "V" Packings, which were originally developed for aircraft hydraulic use. Its impregnation treatment renders it insoluble in, and unaffected by, any type of oil medium. This filler will not wash out.

These packings are usable at temperatures between minus 65° and 180° F., at pressures from zero to as high as 60,000 PSI.

They're long-lived, and are giving excellent service on machine tools, presses, pumps, and other hydraulically operated equipment. For design data, write—

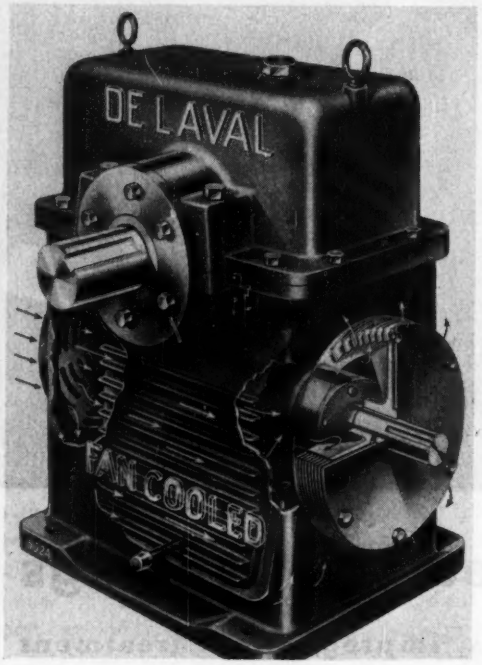
E. F. HOUGHTON & CO.
303 W. Lehigh Ave., Philadelphia 33, Pa.
CHICAGO SAN FRANCISCO DETROIT

HOUGHTON'S *Engineered* **VIM** *Leather Packings*

volts at 5600 revolutions per minute. To meet special requirements, models can be varied.

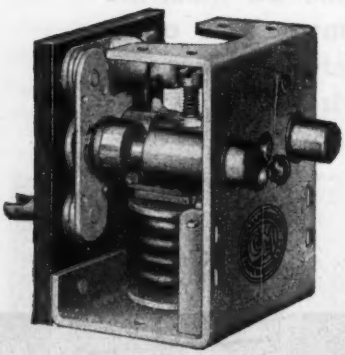
Worm-Gear Speed Reducer

FORCED-AIR COOLING is employed in the new worm-gear speed reducer of De Laval Steam Turbine Co., Trenton, N. J. This cooling removes heat effectively, doubling the capacity of the unit for practically all ratios and sizes at 1750 revolutions per minute. A fan mounted on the worm shaft draws air at high velocity over and



around heat dissipating ribs upon the air side of the oil reservoir housing the gear. Use of a double wall increases effectiveness of cooling by confining the cooling air to the housing and thus securing close air-to-metal contact for maximum heat transfer.

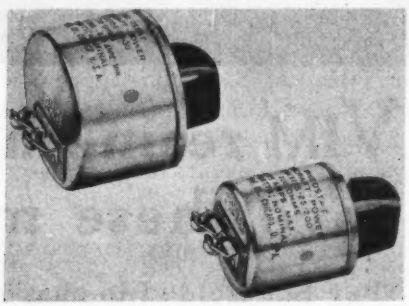
Multiple Coil Circuit Breakers



MULTIPLE COIL overload circuit breakers, introduced by G-M Laboratories Inc., 4300 North Knox avenue, Chicago 41, are designed for protection of either alternating or direct-current circuits. They may be equipped with as many as three separate coil windings, making a single unit adaptable for protection of equipment designed for operation on different voltages. Having no appreciable time lag, they are recommended for use with power vibrators or other electronic equipment to open the circuit instantly when a predetermined overload occurs. Their small size and light weight also makes them suitable for aircraft use or other applications where space or weight is at a premium. Being equipped with independent "on-off" pushbuttons, the breakers can also serve as a manual switch. They are usually factory-adjusted to trip at twice the normal current value with a tolerance of plus or minus 25 per cent, and are not designed for operation on smaller overloads or on closer current tolerances. Contact capacity depends on load characteristics. They have been supplied to handle 6 volts vibrator loads up to 35 amperes normal and with circuit-interrupting capacity to 70 amperes. Contact capacity is 125 grams minimum on each contact. Weighing approximately 9 ounces, the breakers can be supplied with coils and all phenolic parts treated to resist fungus growth.

Aircraft Power Rheostats

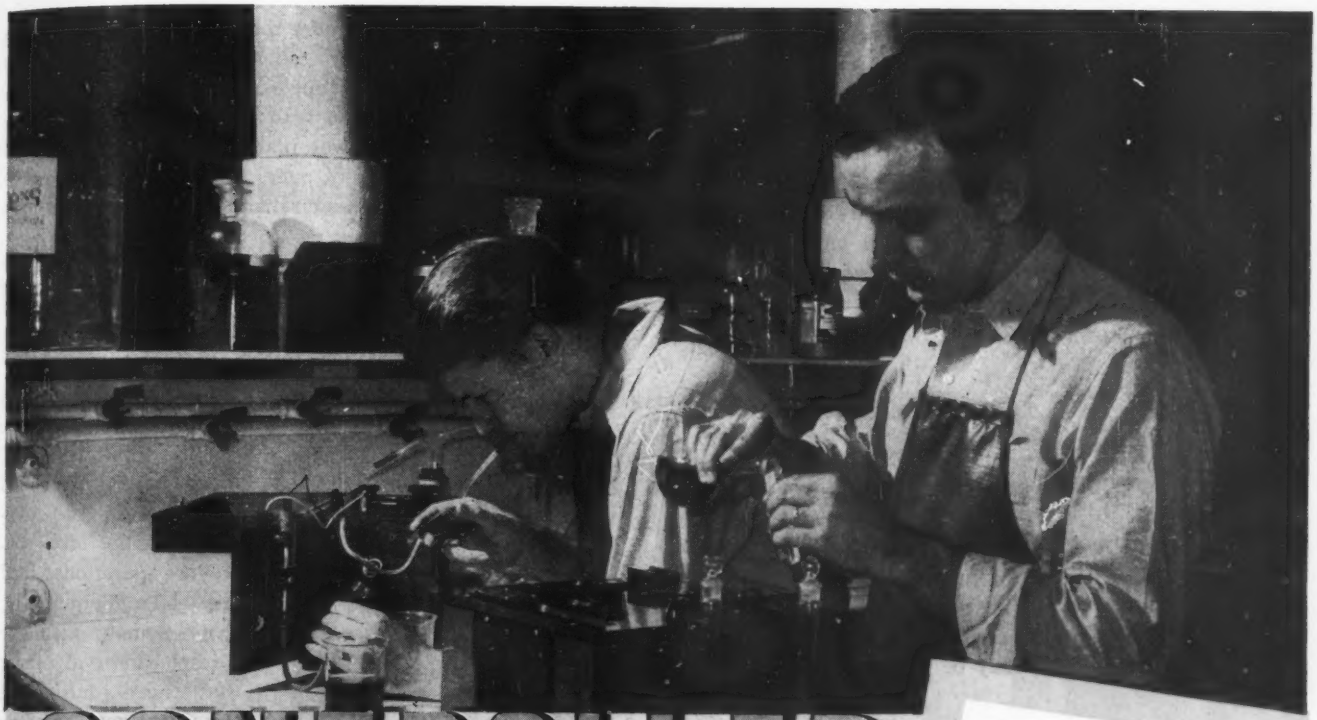
MADE IN ACCORDANCE with the latest Army-Navy Aeronautical Specifications AN-R-14a (Drawing 3155), new power rheostats for aircraft have been announced by The Ohmite Mfg. Co., 4835 Fluornoy street, Chicago 44. In addition to features of other rheostats produced



by the company, the new type has improved control protection. Light in weight—lighter than the allowable weight specified—the rheostats meet the salt-spray corrosion and other critical tests, and operate in the temperature range from −55 to 70 degrees Cent., or −67 to 158 degrees Fahr. Two sizes are available: Model J in 50 watts, and Model H in 25 watts; linear or taper wire-wound, in various resistances, with "off" position as required. Totally enclosed in a compact, corrosion-resisting metal container, the rheostats are rugged in design and construction to provide uniform electrical and mechanical control under operating conditions of cold, heat, humidity, altitude, shock, and vibration.

High Heat-Resistant Resin

DESIGNATED as HM-122 a new molding compound of Lucite methyl methacrylate resin—announced by E. I. du Pont de Nemours & Co. Inc., Wilmington 98, Del.—combines high heat-resistance with other properties previously unobtainable. Its ease and economy in handling and its faster setting properties when properly heated dies are used, contribute to a shorter molding cycle. Because



CONTROLLED QUALITY

one of 8 big reasons why you can depend on Ampco Metal Parts

Quality control is one of the main features that set Ampco apart from ordinary sources of bronze — that assure you of an engineered alloy unvarying in its properties today, next month, or a year from now. This uniformity is important when performance of your equipment depends on material that *always* meets predetermined specifications. • Ampco's laboratory control — a long-established feature of Ampco service — is carried out by the largest corps of technicians in the bronze alloy field.

Ampco also offers you the production "know-how" to *produce* to a standard. • Utilize the exceptional properties of Ampco Metal in your post-war products — it pays! Send us your prints for helpful suggestions. Ask for bulletins.

Only from Ampco can you get *all* of these services and benefits:

1. Distinctive properties of resistance to wear, impact, fatigue, corrosion.
2. A series of engineered alloys—with physical properties to fit your application.
3. Quality control to hold these properties within narrow limits.
4. Diversified production facilities — coordinating in one place all the commonly used metalworking processes.
5. Engineering and production "know-how" to give you a practical manufacturing program.
6. A nation-wide organization of field engineers to assist you.
7. A record of proved performance in hundreds of leading makes of equipment.
8. A national reputation that makes Ampco Metal parts a sales asset.

Specialists in engineering, production, finishing of copper-base alloy parts

AMPCO Metal
The Metal without an Equal

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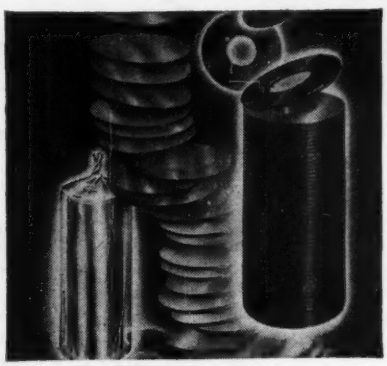
AW-11

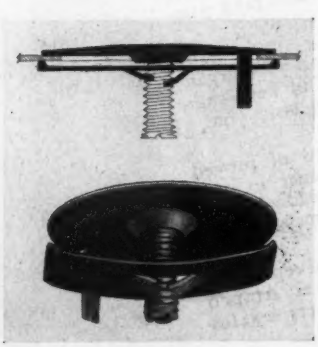
of its resistance to breakdown from heat at molding temperatures, moldings with improved clarity are produced. At the present time the entire output of this material is allocated to essential uses such as colored caps for switchboard light signals, parts for sextants and stethoscopes, blackout lenses for military vehicles, relay box covers, etc. The moldings possess excellent reflecting properties and are free of distortion when molded properly. They are available in a wide range of transparent, translucent and opaque colors. Other characteristics include light weight and toughness.

Carbon Pile Rheostats

MANY NEW TYPES and sizes of continuously adjustable carbon rheostats formed of carbon disk piles have been developed by Stackpole Carbon Co., St. Marys, Pa. By changing the pressure applied to these piles, it is possible to obtain other resistance values within their range without opening the electrical circuits in which they are connected. Pressure may be applied electrically, mechanically, centrifugally or hydraulically. Uses include from both generator and line voltage regulator applications to speed control through governed field current on motors. A carbon disk pile 1½ inches long composed of disks .432-inch in diameter permits a resistance range of from 60 ohms with 1 ounce pressure to .3 ohms at 32 pounds pressure.

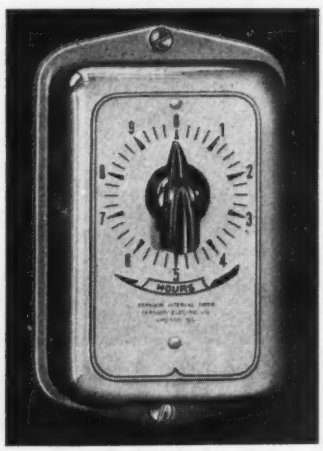


Speed Nut Cover Plates



ORIGINALLY DESIGNED for patching bullet holes in aircraft, the new Speed Nut cover plates of Tinnerman Products Inc., 2085 Fulton road, Cleveland 13, are now being used in industrial applications such as removable covers for inspection doors, access holes and vent holes. They are easily and quickly attached from one side by sliding one end of the Speed Nut into the hole, centering the cover plate over the hole and tightening the screw. The turned-down tab on the cover plate prevents nut from turning while screw is tightened. Made of SAE 1060 steel, heat-treated and coated with zinc chromate primer, the cover plates are available in three sizes to fit a wide range of panel thicknesses and to cover holes 29/32, 1⅛ and 1⅜-inch diameters. Other sizes may be ordered.

Industrial Timer Offered



HAVING WIDE application — in plastic molding, batch mixing, heat treating, enamel baking, liquid agitation, pump and conveyor operations, food cooking equipment, etc.—a new manually preset interval timer has been introduced by Paragon Electric Co., 37 West Van Buren street, Chicago 5. It can be preset to allow a given operation to continue for any predetermined time limit, and to close or open a circuit at the end of the preset time. The timer has a single-pole, double-throw, fully enclosed switch of 1000-watt capacity; two exposed gears, motor pinion and wheel; and a self-starting, slow-speed, synchronous motor. No energy is required of clock motor to trip switch at end of preset time. The time ranges from 0 to 15 seconds through 0 to 20 hours. This 2500 series can be mounted to the surface of any panel or directly to the surface of the equipment the timer is to control. Overall dimensions are 4 7/8 x 3 1/16 x 2 inches.

Precipitation-Hardening Alloy

HAVING EXCEPTIONAL uniformity in response to heat treatment, a new precipitation-hardening alloy has been developed by W. M. Chace Co., Detroit. The chemical analysis is: Copper 60, nickel 20, and manganese 20. It is designated as No. 720 manganese alloy. Although more than half copper, it has a silvery color and presents an attractive surface when polished. This corrosion-resistant alloy is a soft, ductile metal which can be either hot or cold-formed into shapes and then hardened by a low-temperature aging treatment. A hardness of 400 Vickers can be obtained by aging either dead soft or cold-worked material. Physical properties

NO. 1 IN A SERIES EXPLAINING HOW ELECTRONIC TUBES CAN BE USED TO IMPROVE EQUIPMENT DESIGN

Current rectification and motor control by the THYRATRON

Two of the chief applications of electronic tubes in equipment design may be named as (1) to rectify electrical current without the use of rotating equipment, (2) to provide automatic regulation of speeds, temperature, etc. In the thyratron —most versatile of tubes—these functions are efficiently combined.

Thyratron equipment is employed to operate small d-c motors from an a-c power source, and at the same time keep these motors running at the proper speed, regardless of varying loads. The thyratron is a gas-filled tube with one or more grids to control power with split-second precision. Pioneered by General Electric, it performs numerous valuable functions in industry with which design engineers may profitably acquaint themselves.

The many advantages of electronic-tube applications of various types justify a thoroughgoing study of such applications *in every case* where a design is on your boards for development. General Electric will be glad to cooperate in this study, by providing engineering advice on which you may safely base final construction plans. For general or specific information about G-E electronic tubes and their industrial applications, consult your nearest G-E office or distributor. Also ask for the illustrated book on "How Electronic Tubes Work." It is filled with facts about the way tubes operate, how they are classified by design and function, and the many difficult tasks you may turn over to them with confidence. *Electronics Department, General Electric, Schenectady 5, N. Y.*

TUNE IN General Electric's "The World Today" and hear the news from the men who see it happen, every evening except Sunday at 6:45 E.W.T. over CBS network. On Sunday evening listen to the G-E "All Girl Orchestra" at 10 E.W.T. over NBC.

THYRATRON GL-3C23
PRICE $9

This widely used thyratron is a rectifier-regulator, 3-electrode tube of medium output—1,250 v (peak inverse) and 1.5 amp (average). The coated-filament cathode is a quick-heating type; only 15 seconds are required. Exemplifying an important industrial use of the thyratron group, type GL-3C23 is especially applicable to motor control circuits. Singly it can be used for motors of ⅓ to ½ hp, and in pairs for larger motors up to 3 hp where 3 amp is required.

The gas mixture of argon and mercury vapor helps to provide constancy over an exceptionally wide temperature range —from −40 to +80 C—which is important where motors must be started in sub-zero weather or function at high temperature levels. Another feature is uniformity of electrical characteristics. This gives assurance of successful operation to the design engineer who includes the GL-3C23 in his circuits.

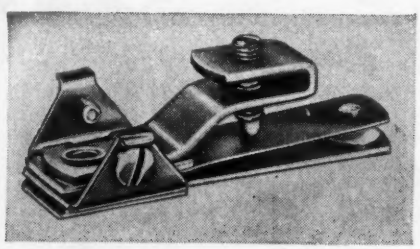
G. E. HAS MADE MORE BASIC ELECTRONIC-TUBE DEVELOPMENTS THAN ANY OTHER MANUFACTURER

GENERAL ELECTRIC

corresponding to this hardness are: Tensile strength, 200,000 pounds per square inch; yield stress for .1 per cent offset, 170,000 pounds per square inch; and proportional limit 120,000 pounds per square inch. Freedom from "drift" and high fatigue strength make the alloy especially applicable as a spring material. It is available in all forms in which stainless steel is supplied. Cold-rolled strip is produced to close thickness tolerances for precision spring applications.

Built-in Type Thermostat

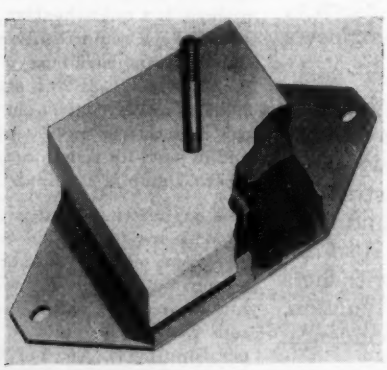
CLOSER AND QUICKER adjustments are possible with a new feature incorporated in the line of SK thermostats and thermal timers of George Ulanet Co., 88 East Kinney street, Newark 5, N. J. No lock nut is necessary in the new design. The adjustment screw passes through



two collinear tapped holes on a U-bracket. This U-bend imparts a spring action on the thread of the adjustment and prevents its turning by any but manual means. Thus a closer adjustment is possible.

Vibration Mounting Available

DESIGNERS CONCERNED with eliminating vibration and excessive noise will be interested in the new Rexon vibration mounting introduced by Hamilton Kent Mfg. Co., a unit of U. S. Stoneware, Akron, O. This mounting utilizes the vibration dampening properties of rubber in shear but with no possibility of overloading the shear

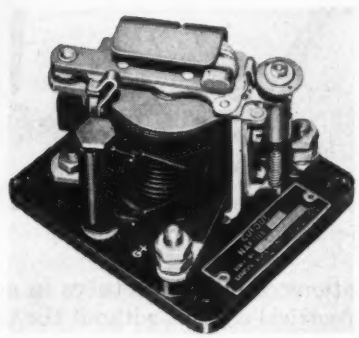


elements. This property stems from the use of a special "X" type design of the rubber element which acts in shear at the points of the "X" under normal loading, but in compression when load exceeds the rated capacity of the mounting. The mountings, it is claimed, cannot be damaged by temporary or permanent overloads, and are easily installed without use of special angles, brackets or radius rods, and without increasing the overall height of the machine more than 1½ to 2 inches.

Flexible Metallic Packing

CAPABLE OF withstanding temperatures to 2000 degrees Fahr. and resistant to corrosive gases, alkalies and most acids, a new type of flexible metallic packing has been announced by Johns-Manville, 22 East Fortieth street, New York 16. Known as the Inconel packing, its construction and toughness make it adaptable to many industrial uses, and will stand up under the conditions existing in exhaust systems of airplane engines equipped with turbo-superchargers. Made of nickel-chromium-iron alloy in wire form, which is first knitted into mesh and then braided, the new packing is strong, resilient and nonscaling and nonmagnetic. It is available in a number of styles, including square section packing in coil and tape forms, from which ring type gaskets may be formed. Another packing is made with asbestos fibers interwoven with the metallic alloy, serving as a sealing agent.

Reverse-Current Cutouts



DESIGNED TO PREVENT reverse flow of current to generators in aircraft power systems utilizing batteries and generators as a power source, reverse-current cutouts are now being offered by the Eclipse-Pioneer Division, Bendix Aviation Corp., Teterboro, N. J. They provide a lightweight means of automatically disconnecting generators from electrical systems when generator voltage drops below system voltage. The cutouts can also operate as generator automatic main switches. Available in four types in various ratings, the cutouts operate satisfactorily in any mounting position. Changes in temperature have a minimum effect on their operation.

High Heat-Resisting Paint

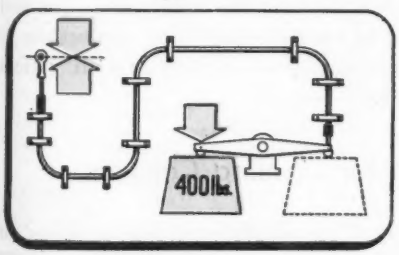
PRIMARILY FOR use on surfaces subjected to high heat, Triple-A paint announced by Quigley Co. Inc., 527 Fifth avenue, New York 17, offers maximum protection and decoration. Metal surfaces to be painted should be clean and dry, free from heavy rust, heavy mill scale, grease, dust, or other paint or organic matter. The paint, which is furnished in the proper consistency for use, is applied in a thin coat, well brushed out. It is then air-dried for 1½ hours. For interior use where moisture conditions are normal the paint is ready for service, while where ex-

Dependable *all around duty assured with* Simmonds Push-Pull Controls

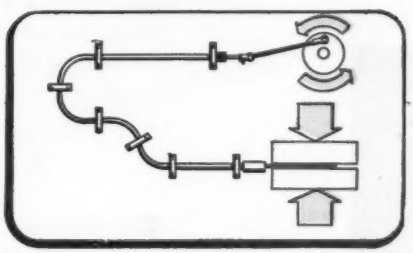
Read how these other tests, conducted to AAF specifications, show how Simmonds products more than qualify for EXTRA DUTY as well as NORMAL DUTY operations.

FIRST TO CARRY THE YELLOW DOT

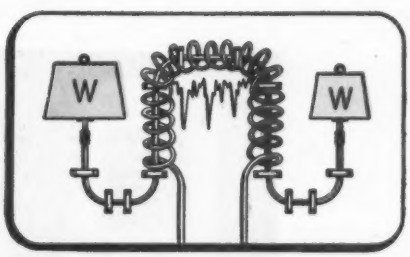
of Army Air Force Winterization acceptance for operation in extreme temperatures is the Simmonds-Corsey Push-Pull Control. Comprehensive tests with temperatures as low as minus 72°F. and as high as 160°F. prove efficient operation under wide variety of conditions.



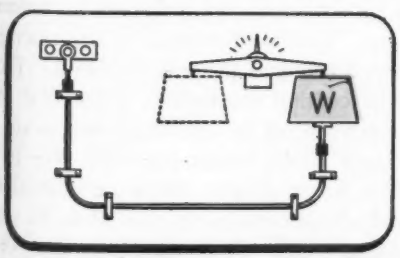
STATIC STRENGTH: *In this test Simmonds push-pull control successfully withstands tensile and compressive forces of 400 lbs. Credit simplification of design, improved linkage.*



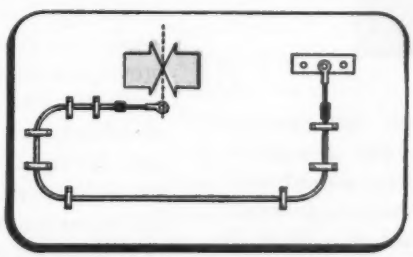
ENDURANCE: *Cycled 30,000 times under stress, as illustrated, Simmonds controls are not affected in any way, and can be expected to outlast the life of unit served.*



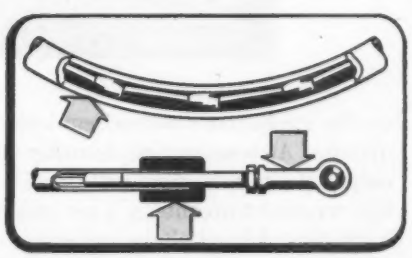
EFFICIENCY: *Input-output ratio of control is measured at various temperatures in insulated chamber. Result: average efficiencies are twice the AAF requirements.*



PRECISION CONTROL: *Motion between tension and compression is negligible. Before endurance test: average—.046"; after—.082". Simmonds are precision-built controls.*



DEFORMATION: *AAF specifications call for loads ranging from 10 to 50 lbs. Allowable average deformation is .140". Simmonds controls average only .083".*



CORROSION: *Simmonds controls meet AAF specifications for corrosion resistance with standard cadmium plated or anodized surfaces. Tube ends are rubber sealed.*

INQUIRIES concerning War Contracts or Post-War problems involving push-pull control equipment are invited. Our service engineers will furnish you gladly with analyses and recommendations. Telephone or write to your nearest Simmonds office.

SIMMONDS EQUIPMENT FLIES WITH EVERY TYPE OF ALLIED AIRCRAFT

Automatic Engine Controls — Push-Pull Controls — Spark Plugs
Hydraulic Accumulators — Hydraulic Fuses — Chronometric Radiosondes
Self-Aligning Rod-End Bearings — Fasteners and Clips of Specialized Design

30 Rockefeller Plaza,

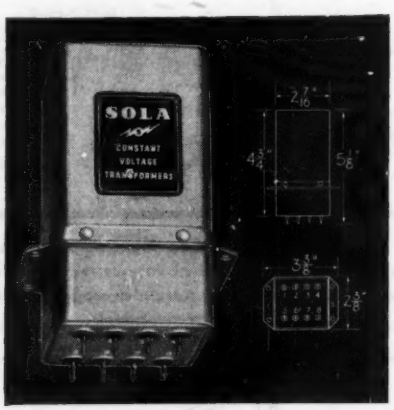
SIMMONDS AEROCESSORIES INC.

New York 20, New York

tremely wet conditions prevail, it is necessary to heat the paint film after application to produce a coating that will not be affected by moisture and render the film insoluble. Under these conditions the paint film should be heated to at least 300 degrees Fahr. When the paint is used on exterior surfaces, a heat cure of 500 degrees Fahr. is necessary to obtain maximum resistance to weather. For surfaces continually hot no heat treating is necessary—whether these surfaces are exposed to weather or interior room exposures. While it is preferable to apply coating at normal temperatures, the paint can also be sprayed on hot metal surfaces. Resistance to heat depends on the surface to which it is applied. On light steel, it will adhere under rapid heating and cooling up to 1400 degrees Fahr., on alloy steel, brick, etc., it will stand 2500 degrees Fahr. or over. The paint is nonflammable, non-irritating, and does not give off fumes when applied, dried or when subjected to heat or flame.

Constant Voltage Transformers

HERMETICALLY SEALED constant-voltage transformers for through-chassis mounting have been announced by Sola Electric Co., 2525 Clybourn avenue, Chicago 14. These new transformers are applicable to heating and refrigeration controls, television and FM receivers, vacuum tube voltmeters, electronic gaging and inspection equipment, photometric instruments and other



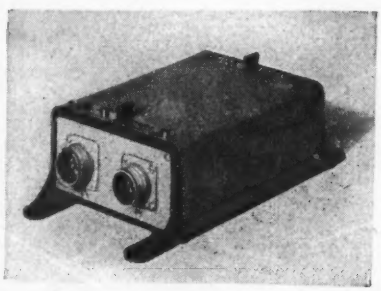
similar applications where regulated supply voltage is required. At a reasonable transformation ratio of input to output the transformers will provide a single output voltage constant to within ±1 per cent of rated requirements regardless of line voltage variations as great as ±12 to 15 per cent. The units are available in capacities up to 15 volt-amperes, 60 cycle operation, and are supplied with a separate capacitor for external mounting.

Fastening Offered in Brass

RECENTLY ANNOUNCED by The B. F. Goodrich Co., Akron, O., the Rivnut blind fastener, which serves as a nut-plate, rivet or both, is now being made of brass as well as aluminum. Tensile and shear strengths of the fastening made with the brass alloy have been improved considerably. Originally made of aluminum for aircraft applications, the fastener is being adapted by other industries which called for use of strong metals. The development of the brass Rivnut is a first step in that direction.

Dust-Tight Enclosure

FOR HOUSING combinations of aircraft control devices, a new standard dust-tight enclosure has been announced by the Industrial Control division of General Electric Co. The enclosure accommodates various aircraft control relays and contactors with ratings up to and including the G-E 100-ampere aircraft contactor. Completely gasketed, the enclosure is made of aluminum, with



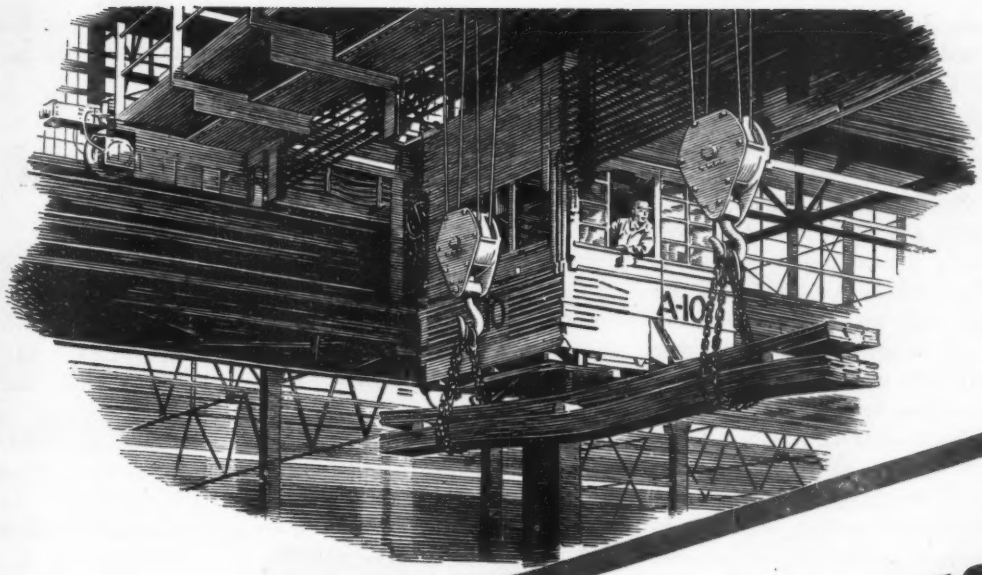
cast aluminum end frames. A self-locking latch permits removal of the cover of the enclosure in a vertical direction without disconnecting wires or the use of tools, and has a provision for wire-locking the enclosure. The end plates are adaptable to various sizes "AN" connectors or terminal posts. This enclosure is furnished in various lengths up to 19 inches, over-all.

Process for Coating Copper

PROBLEMS IN THE preparation of copper alloys for organic finishing have been eased by the new Ebonol "C" process of the Enthone Co., Elm street, New Haven, Conn. By this method an adherent, nonreactive cupric oxide coating on copper alloys is produced that gives adhesion to lacquers, paints and enamels under severe weathering conditions. The inert nature of the coating prevents reaction between it and the organic finish. The finish obtained by the coating is nap-like, presenting an absorbent base for the paint to be anchored, and is suitable for treating copper alloys containing from 60—100 per cent copper. The lower copper alloys are colored a mahogany brown and alloys containing more than 65 per cent copper are colored black. It is, therefore, not suitable as a base for clear lacquers or for one coating of white pigmented lacquer due to its dark color. Requiring a treating time of approximately 10 minutes, the finish is applied by immersion of the work in the solution operated near 210 degrees Fahr.

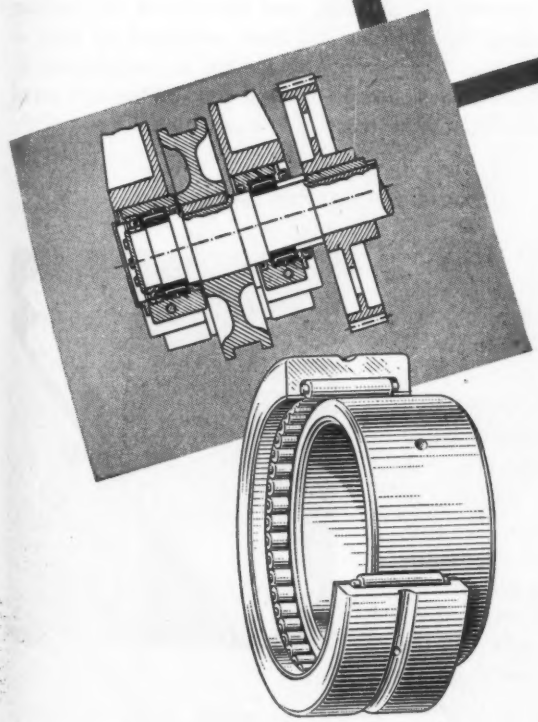
Hose Clamp Announced

AVAILABLE FROM the Punch-Lock Co., 321 North Justine street, Chicago 7, the new Punch-Lok hose clamp is furnished in diameters from ¾ to 48 inches for connecting various kinds of fittings, special nipples, or ordi-



Drawing from photograph furnished by Carnegie-Illinois Steel Corporation

PUTTING TONS ON THE SPOT



Whether it's a lifting job that calls for the maneuverability and operating ease of a lift truck, or a multi-ton lift that demands the strength of a giant traveling crane—*putting tons on the spot* is done by effortless, finger-tip control, when loads travel on anti-friction Torrington Bearings.

In the application illustrated, the high load capacity and compact design of the Torrington Type NCS Heavy-Duty Needle Bearings make them ideal for installation in the crane's axle housings.

Out of the ordinary bearing applications are an important part of the service of Torrington's Bantam Bearings Division. Whether you need counsel on standard or large, custom-built anti-friction bearings, consult our engineering staff.

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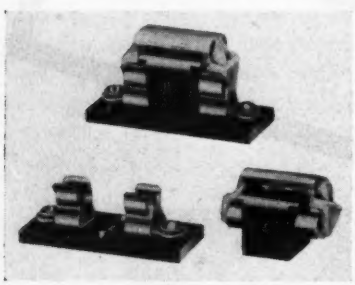
nary pipe to hose. All hose, from high-pressure wire-woven and braided hose used in controls of hydraulic machines, or any flexible coupling between rigid pipe lines or fittings for air, steam, etc., to ordinary air and water hose, may be fitted with this clamp. In making the connection, a broad, flat, high-tensile-strength, galvanized-steel band is double-wrapped around the joint. After tensioning with a pull of 1000 pounds within the locking tool, the ends are securely locked together without loss of tension within the flat pressed-steel clip, and the excess band is cut off, flush with the clip. When the clamp is locked, vibration or rough handling will not loosen it. The No. 3 clamp with a ⅜-inch band width is designed for applications ranging from ¾ to 3/16-inch diameter hose. Clamps ranging in diameter from 1 to 48 inches have a band width of ⅝-inch. For electrical applications or for splicing flexible welding cable where corrosion resistance is required, the clamps are available in all sizes in a copper base alloy, in preformed or open-end type. The latter for lacing around the object where end obstructions would prevent use of preformed clamp.

Small, Light Circuit Breaker



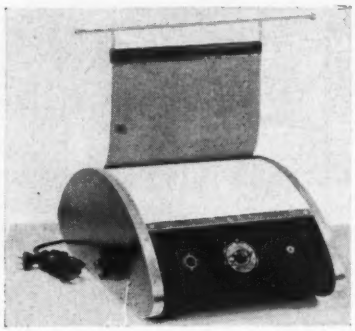
TERMED THE "Breakerette", the new No. 1561 circuit breaker of Littelfuse Inc., 200 Ong street, El Monte, Calif. is interchangeable with all 5 A G fuses or Navy "midget" size. (The 5 A G fuses are 1½ inches x 13/32-inch diameter, and from 3 to 50 amperes.) The new circuit breaker is rated at 32 volts alternating or direct current, with the snap action capable of interrupting short circuits of 100 amperes in ratings up to 5 amperes and 2500 amperes in ratings over 5 amperes capacity. Time characteristics of the fuses are duplicated electrically. In operation when the bimetal releases the pushbutton on which it acts directly, a double break of the circuit is effected. Actual breaking distance is 5/16-inch. To reset the breaker, the button is pushed into its bottom position. Both trip-free and nontrip-free features are provided. The case of the small and light Breakerette is molded bakelite, while the trip-free shield is thermoplastic to permit visual indication. Extreme dimensions are 1⅞ inches x ¾-inch. Overall height with the trip-free shield is 1¼ inches. Weight without the hood is 15 grams; while with the hood, the weight is 18 grams.

Fuel-Resistant Material

DEVELOPED BY Los Angeles Standard Rubber Inc., 1500 East Gage avenue, Los Angeles, a new Buna N fuel resistant stock (233-5) retains a tensile strength of over 1000 pounds per square inch and an elongation of over 100 per cent—this after being subjected to aircraft engine oil at 300 degrees Fahr. for a period of 70 hours. The material was developed for oil seals, oil rings and other applications requiring resistance to aromatic fuels. It remains flexible at 40 degrees below zero, and has a low compression set.

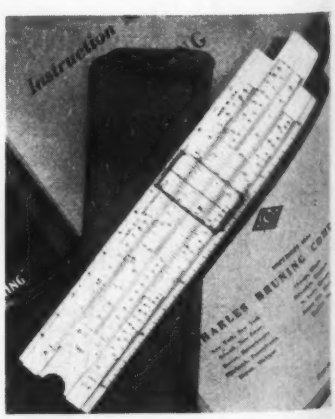
Engineering Dept. Equipment

Printing Machine Announced



LATEST ADDITION to the line of electro-copyist machines produced by Hunter Electro-Copyist Inc., Duquid building, Syracuse 2, N. Y., is its Model No. 50. With a capacity of 12 x 18 inches, the machine has a width of 14¾ inches, length 24½ inches and height 8½ inches. Weight is approximately 25 pounds. The exterior, of seasoned gum surface plywood, is walnut stained and lacquered, with a bright chrome trim. Lighting is furnished by four 60-watt and six 40-watt white lamps. The 60-watt lamps are used for making negatives, and both the 60 and 40-watt lamps may be used for positive prints. A toggle-switch controls the lighting, and the period of exposure is controlled by a mechanical timer mounted on front of the machine. Even pressure surface is maintained by tension springs attached to a metal rod, pulled into metal hooks attached to the front of the machine.

Precision Pocket Slide Rule



NOT A "DURATION substitute" but rather a precision instrument, a 5-inch pocket slide rule is being offered by Charles Bruning Co., New York. A notable feature of this rule is the precision of its graduations which are molded in as an integral part of the rule. To facilitate reading, graduations and numerals of the CI scale are in red. Three screws in the back provide a single adjustment for tension on the slide. The glass indicator, which is easily replaceable, is enclosed in a frame of steel that holds it in place. A, B, CI, C, D, K, S, L and T scales are shown on the rule in order to adapt it to a wide range of service. The beveled edges of the rule are in graduated scales of both inches and centimeters.

J&L STEEL

Our metallurgical engineers will be glad to discuss your production problems with you.

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SPECIAL COLD DRAWN SHAPES SAVE MACHINING

David Sussin

Dr. Mervin J. Kelly

Jacob Bertschi

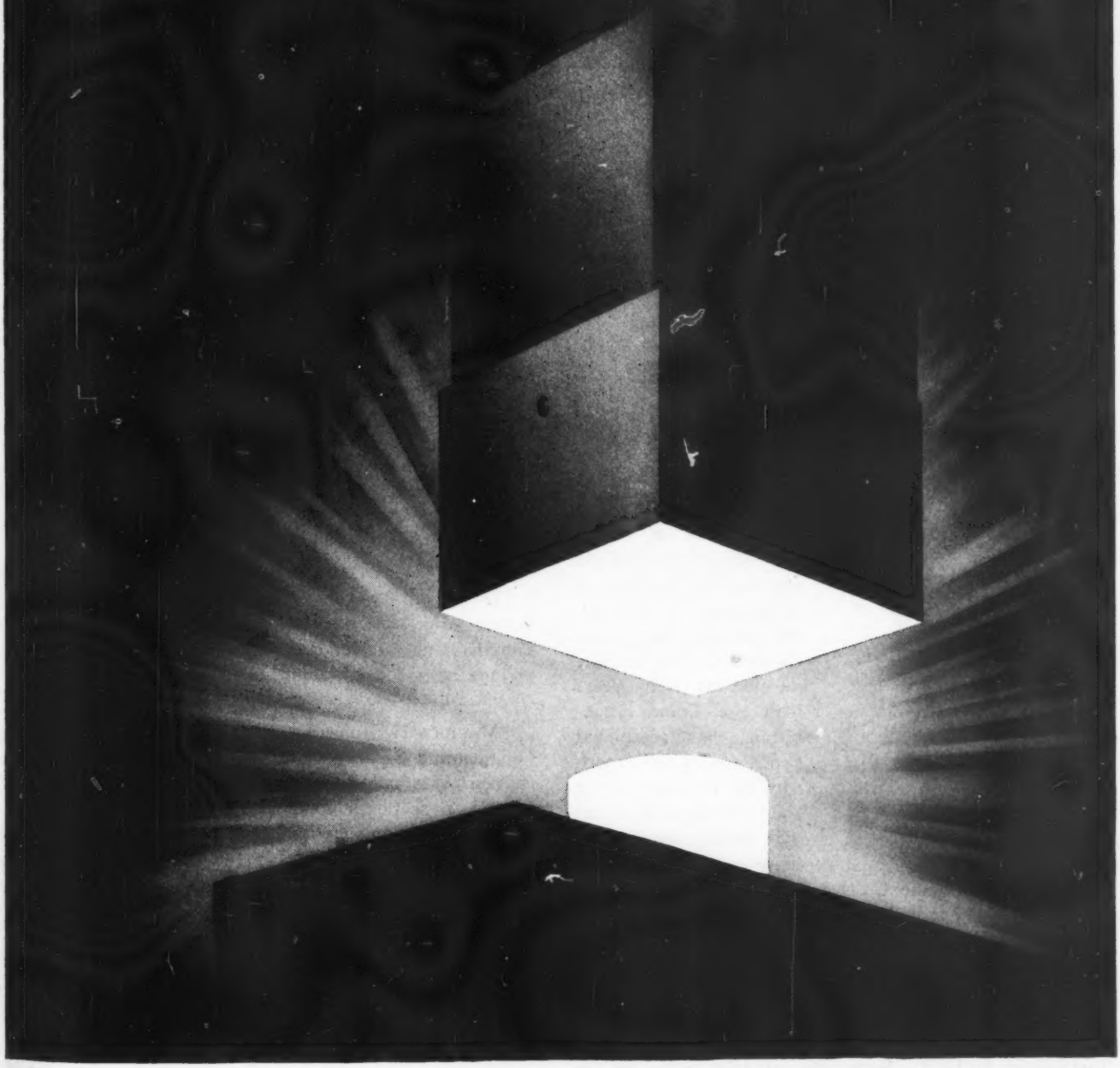
MEN...of...machines

DAVID SUSSIN, in his newly created position as chief of research for Kelley-Koett Mfg. Co., has been given the responsibility to forecast the future anticipated expansion applicable to X-ray and allied applied science. He will devote his efforts to continuing research on engineering, medical and industrial techniques to the end that new inventions and innovations broadening the scope of X-ray for medical and industrial applications can be developed and utilized. Mr. Sussin came "from the ground up" with the Kelley-Koett company until he rose to the position of chief engineer, which post he held prior to his present appointment. He obtained his bachelor's and master's degrees in electrical engineering at the University of Texas, and later while working for his doctor's degree in mathematics and physics at the university was an assistant on the faculty.

MERVIN J. KELLY, director of research of Bell Telephone Laboratories, New York, has been elected executive vice president of the laboratories. A scientist of distinction, Dr. Kelly has made many improvements in photoelectric cells for sound pictures, in watercooled tubes used for high-power broadcasting, and in the vacuum tubes for long distance telephony. A graduate of the University of Missouri in 1914, Dr. Kelly received his doctor of philosophy degree from Chicago in 1918. In the same year he entered the Bell Laboratories as a research physicist, and in 1930 became vacuum-tube development director. In 1936 he was made director of research. He has been in charge of Bell Laboratories' development in the radar and other important fields, and is an advisor to the government on technical war projects.

JACOB BERTSCHI has been appointed chief designer for Carter Motor Co., Chicago, in recognition of his long service in the trade and the commercial significance of his developments in the field of rotary power units. Particularly well known for his work in the permanent-magnet dynamotor field, and mostly responsible for the company's MagMotor line, Mr. Bertschi recently developed the smallest multi-output dynamotor capable of handling any appreciable power. He is also credited with the development of the hand generator, many of which

Molybdenum die steels serve particularly well where heavy dies require deep hardening.



CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

ARMY E NAVY

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Climax Molybdenum Company

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MOLY

are seeing active war duty now and which will power peacetime expeditionary communications equipment. Born in Switzerland, Mr. Bertschi came to America in 1908 and entered the radio parts design field in 1923. From then on he had been active in this industry until he became associated with Carter Motor Co. in 1942. Essentially a design executive, Mr. Bertschi is known mostly for his intimate knowledge of metals and their uses.

♦

GEORGE A. MUELLER has been made chief engineer and assistant factory manager of Staples Tool & Engineering Co., Cincinnati.

♦

IRVING I. FELDMAN, formerly assistant chief designer of Product Engineering Service Co., Detroit, is now a partner of Aero-Automobile Engineering, of the same city.

♦

PHILLIP ALBANESE has become connected with Aerojet Engineering Corp., Pasadena, Calif., as development engineer. Formerly he had been project test engineer for Dodge Chicago Plant, Division of Chrysler Corp.

♦

WILLIAM GOODMAN, formerly consulting engineer for the Trane Co., has joined the staff of the Illinois Institute of Technology as professor of mechanical engineering. Mr. Goodman, who was associated for twelve years with the Trane organization, has completed two decades of experience in air conditioning and refrigeration, and will handle courses in these and related fields. While with the Trane Co. he designed air conditioning equipment and handled special problems in the application of the company's products to air conditioning systems. He also carried on extensive research in heat transfer problems and wrote several books on air conditioning. Mr. Goodman received his bachelor of science degree in mechanical engineering from the Armour Institute of Technology in 1942. Soon after graduation he got his start in air conditioning by becoming associated with a consulting engineer and later working with other Chicago engineering firms. He then worked as an independent consulting engineer on various types of mechanical and electrical equipment.

♦

CAMERON N. LUSTY, who had been chief engineer for Lloyd Mfg. Co., Heywood-Wakefield Co., and Mercury Aircraft Co., Menominee, Mich., has joined the Frankfort Sailplane division of Globe Corp., Joliet, Ill., as development engineer.

♦

ELMER RAYMOND UR has been appointed design engineer for Jacobs Aircraft Engine Co., Pottstown, Pa. He had been aircraft engine designer for Pratt & Whitney Aircraft Division, United Aircraft Corp.

♦

H. A. STORCH, former design and development engineer at Harris Products Co., Cleveland, has been made project engineer of Evans Products Co., Detroit.

♦

CHARLES W. VAN OVERBEKE has resigned as an instructor in mechanical engineering at Purdue university to accept the position of experimental engineer for Ex-Cell-O Corp., Detroit.

♦

JOHN R. QUINZIO left his post as designer with Sterling Engine Co., Buffalo, N. Y., to become connected with Buffalo Foundry & Machine Co.

♦

THOMAS T. SCHROETER has been appointed mechanical engineer for the National Advisory Committee for Aeronautics, Cleveland.

♦

C. P. CROCO, formerly director of engineering for the Federal Machine & Welder Co., Warren, O., has returned to Westinghouse Electric & Mfg. Co. as manager of the recently formed welding department of the motor division at the Trafford, Pa. works of the company. He will assume responsibility for engineering, manufacture and sales of welding equipment manufactured at this plant.

♦

DAVID J. LITTLE, who had formerly done aeronautical engineering work in Detroit, has recently become aeronautical engineer for Hartwig Aircraft, San Antonio, Tex., where he will manufacture helicopters of his own design.

♦

MAURICE H. HOBBS has been appointed manager of the engineering department of the switchgear and control division, Westinghouse Electric & Mfg. Co. CHARLES P. WEST, a section engineer, has been made manager of switchboard engineering, while WILBUR C. FULTON, switchboard engineer, becomes section engineer in switchboard engineering.

♦

JOHN OTTO ALMEN, head of the mechanical department No. 1, General Motors Research Laboratories, was recently awarded the Manly Memorial medal by the Society of Automotive Engineers for his work in developing methods for increasing working strength of metals.

♦

S. L. EASTMAN, who has been connected with Cleveland Worm & Gear Co. since 1927, has been appointed chief engineer of the company. Since April, 1942, Mr. Eastman had been assistant chief engineer.

♦

DR. ALEXANDER J. ALLEN has been made Westinghouse Graduate Professor of Engineering, University of Pittsburgh.

♦

PAUL M. CHRISTENSEN has been added to the staff of engineers at The Clark Controller Co. Mr. Christensen had spent eighteen years with another electrical control manufacturer, lately as chief engineer in charge of development and design.

FORT PITT
TUFALLOY
STEEL CASTINGS
PORTER
"Better Built"
Equipment
Established 1866
High Strength, High Elongation, Shock Resistant and Tough!
This versatile family of electric furnace alloys provides a remarkably wide range of higher physical properties required to meet the extreme conditions of modern processing. With resistance to shock, and other outstanding dynamic properties plus good machinability, TUFALLOY steel castings are adaptable to many uses in the design of process equipment. Fort Pitt metallurgists and engineers will be glad to work with you in selecting an analysis best suited to your requirements. Send us the blueprints and we will submit estimates.
MT. VERNON CAR DIVISION:
Complete Line of Freight Cars
LOCOMOTIVE DIVISION:
Diesel, Diesel-Electric, Electric, Steam, and Fireless Steam Locomotives.
PROCESS EQUIPMENT DIVISION:
Complete Line of Chemical, Food, and Petroleum Refinery Equipment.
QUIMBY PUMP DIVISION:
Screw, Rotex, Centrifugal, Chemical Pumps.
STEEL CASTING & FORGING DIVISION:
Electric Furnace Alloy, Stainless, Heat-Treated and Pressure Steel Castings, Forgings.
Fort Pitt Steel Casting Division
H. K. PORTER COMPANY, Inc.
PITTSBURGH 22, PENNSYLVANIA
Factories: McKeesport, Pa. • Pittsburgh, Pa. • Blairsville, Pa. • Newark, N. J. • New Brunswick, N. J. • Mt. Vernon, Ill.

Weldment Design

(*Concluded from Page* 142)

this portion of the weldment would be more costly in plate. *Fig.* 18 is the weldment in which the previously shown casting is used. Six such castings or larger ones are used in this one piece.

There is a tendency to utilize standard structural rolled shapes in machinery weldments though their value as components is questionable in many instances. This is logical when it is remembered that available rolled shapes were evolved for an utterly foreign purpose. Hence their use in machinery parts often seems a makeshift. *Fig.* 19 is a typical elementary comparison. The original design—utilizing I-beams and angles—is shown on the left with the redesign adopted on the right. Welding is reduced, lateral stability is increased, and the appearance is better. Also, abrupt projections are eliminated, fitting and welding are simplified and the transverse member is more effective at equal weight.

Special Shapes for Weldments

It is possible that shape mills will evolve new sections as required tonnage develops and if so, the shapes will be designed to suit this new function. We operate a small tonnage mill which provides rough shapes such as these shown in *Fig.* 20, solely for weldments. Roll cost is low but this is essential to the relatively small tonnages required. As a result, surface finish is not equal to that of the usual shape in quality but the product is well suited to its use. This mill has provided added design freedom

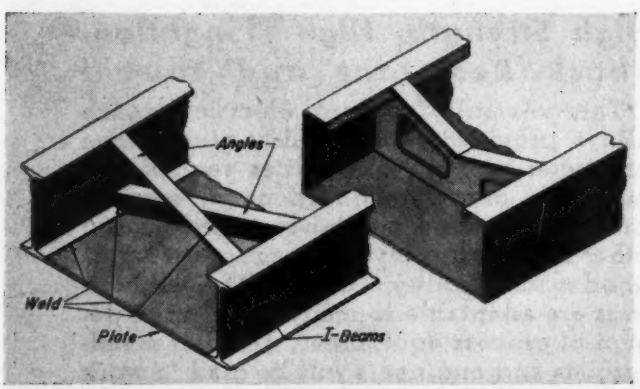

Fig. 19—Above—Comparison shows design utilizing I-beams and angles, left, and simplified design at right

through the use of such components, which formerly were machined from solid slabs.

Finally, there is the important subject of specifications. No well-defined trend is as yet developing in the specification of welded machinery parts, except that progress has occurred in nomenclature. Weldments procured by or for government agencies, for instance, are clearly specified and type, extent and control of inspection are fairly well developed. On the other hand, specifications on commercial welded machinery parts are too often pretty general and lean, and this causes discouraging delays in pricing and final design. It would seem that valuable and important work can be done by a coordinating agency since there is splendid opportunity for the compilation and correlation of design criteria.

As mentioned, nomenclature is comparatively well developed and welding symbols are standardized. But many items of specification on a welded machinery part are somewhat complex in contrast to those on a casting. For example:

What type of weld joint shall be used and where? (Much remains to be done in the evaluation of types of joints before the basis of this specification is available).
What type of electrodes shall be used and where?
What method of inspection shall be used and where?
What are the standards for such inspection?
How and to what degree shall the weldment be tested?
Should it be stress relieved?
What tolerance can be expected and where?

Today the codes and standards of many agencies overlap in the specification of welded machinery parts. It is reasonable to assume that none is adequate, for they were

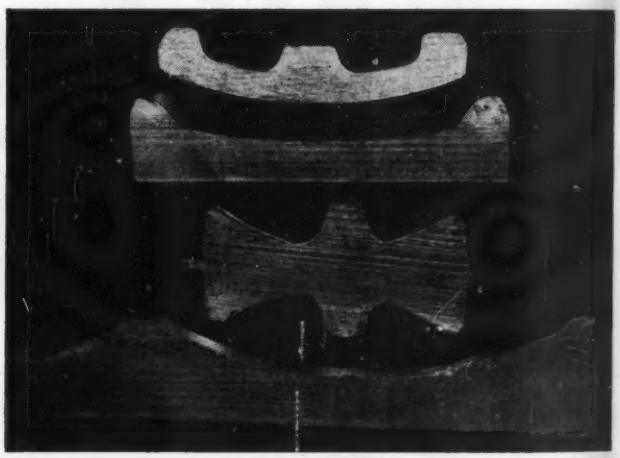

Fig. 20—Above—Rough shapes such as these are rolled for weldments, giving added design freedom for such components which formerly were machined from solid slabs

established for other purposes. My conviction is that the commercial designer literally has no guiding standards with the exception of nomenclature. It is conceivable that he does at times apply weldments in which faulty design endangers human life. Common procedure is for the user to rely on the supplier to develop his specifications for him, but there is nothing alarming in this if fundamental standards are available.

Specifications undoubtedly are guided often by equipment limitations, inventories, geographical location, individual experience or the lack of it, or ill-advised convictions. All these can generate a rather chaotic, unstabilized condition in the industry, assuming that the production of welded machinery parts properly can be termed an industry. Costs and prices are widely divergent and will so remain until fundamental specification standards are devised and promulgated.

Much experience and basic information with service records as a sound foundation exist. Ultimately valuable work will be done in its compilation, correlation and in filling of the gaps.

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ASSETS to a BOOKCASE

Cast Metals Handbook

Published by American Foundrymen's Association, Chicago; third edition, 765 pages, 6 by 9 inches, flexible clothbound, available through MACHINE DESIGN, $6.00 *postpaid.*

It is well to recognize that the high speeds, ultra precision, and unusual stamina of modern machines, all achieved within minimum space and weight requirements, are in great measure due to the development and competent use of better engineering materials. Selecting the material most suitable for a given machine part, and designing the part to take full advantage of the properties and characteristics of the material specified, can be accomplished only when the designer really understands materials.

While this thorough understanding of materials is highly desirable, not everyone has the time—nor for that matter the inclination—to attend engineering-society lectures or to ferret from scattered literature the latest information available. This being the case, the most logical alternative is for someone or some organization to do the ferreting and correlating, and to present the most pertinent and useful data in the handiest possible form.

This is precisely what the American Foundrymen's Association has done as regards cast metals in this splendid book which is both a fine text and an authoritative reference. Its early chapters deal with pattern equipment, considerations in casting design, cost estimating, significance of strength and ductility tests, and metallurgy. There follows a particularly valuable chapter dealing with recommendations to designers of steel castings in which the key factors influencing design are comprehensively presented. The remainder of the book is devoted to complete engineering data on all kinds of cast metals, with separate and extensive sections dealing with steel, malleable iron, cast iron and nonferrous alloys. Typical applications for the various metals, data on machinability, heat treatments, corrosion resistance, etc., are given throughout.

□ □ □

Machine Designer's Guide

By K. W. Najder; published by Machine Designer's Guide, Lake Orion, Mich.; 281 pages, 5¼ by 8 5/16 inches, offset printed, clothbound; available through MACHINE DESIGN, $2.75 *postpaid.*

This book is primarily a simple and practical reference of engineering formulas. It places before the designer, under one cover, most of the formulas which have a direct bearing on his work.

Of course the mere presentation of formulas is far from the complete story needed to determine and evaluate stresses and strains, power losses, accelerations, and the like. It is one thing to be able to handle the mathematics involved in utilizing formulas and quite another to know how to apply the formulas competently in design. To assist in this direction, numerous examples are presented to show applications of formulas to engineering problems.

Thus, while this book is not a comprehensive treatise on engineering, it does make accessible, in what might be termed "finger-tip fashion", the basic formulas which most practising engineers at one time learned but cannot readily recall.

□ □ □

Electronics Today and Tomorow

By John Mills; published by D. Van Nostrand Company Inc., New York; 178 pages, 5⅛ by 7¾ inches, clothbound; available through MACHINE DESIGN, $2.25 *postpaid.*

If you are not grounded in the basic principles of electronics and wish to strike up a speaking acquaintanceship with them, this book should help to clear away the veil of mysticism and prepare you for a rational, more extensive study should you so desire.

It has been said that we fear and view with awe only those things we do not understand. All the ballyhoo of recent years has tended to envelop "electronics" with an aura of magic, to an extent that many have come to assume it a science so complex that only a doctor of philosophy can comprehend it. This is not the case. The principles of electronics stem from physics as do the principles of electrical engineering.

This book might be said to take the reader by the hand and lead him step by step through the maze of atoms, ions, electrons, protons, neutrons, cathodes, anodes, grids, photocells and the like, stopping at intervals along the way for introductions and informative discussions. However, it is not in any sense a technical treatise. It contains no mathematics and thus cannot be recommended for those interested in advanced phases of the subject.

Published recently by the Seamless Steel Tube Institute, an elaborate leather-covered, loose-leaf bound book, "Seamless Steel Tube Data", gives just about all of the information pertaining to such tubing that an engineer requires. The data presented on mechanical and pressure tubing have been compiled through the collaborative efforts of many leading producers. In addition to these data on available forms, sizes and costs, numerous formulas useful in calculations attendant to the use of tubing are supplied. It is a well-prepared volume, amply illustrated, conveniently indexed, and available at $2.50 from Seamless Steel Tube Institute, Gulf Bldg., Pittsburgh.

"The POLLAK Version of Re-conversion"

Includes the Pooling of Development and Designing Experience

This Version of Reconversion does not include delving into the dream world for new product designs but rather approaches the problem from a practical, down-to-earth standpoint. The revolution in materials and product design when peace comes, may not be as fanciful as some reports have stated but the lessons in development and design for war should be utilized for improving future consumer products.

This plan includes necessary industrial research and production development to obtain facts and point out efficient and economical procedure—then comes the blueprint stage, followed by working models.

Working in this manner with the sales and engineering staffs of American Manufacturers, POLLAK has already developed some new and needed products, which are awaiting V-Day for the start of production.

POLLAK is ready to combine the development and designing knowledge and experience of its engineers and technicians with your planning staffs—to save time later in the reconversion periods. Consultation is invited without obligation.

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Design Abstracts

How Heat Affects Rubbers

MEASUREMENTS of the relaxation of stress in thin rubber bands held at constant elongation, and measurement of the creep of bands supporting a constant load, have furnished interesting and valuable information about the deterioration of rubber at high temperatures. It has been found that in both natural and synthetic rubbers two chemical reactions take place simultaneously at high temperatures, changing the structure of the rubber network. The first is a reaction which cuts the long chain molecules in the rubber, and the other is a reaction in which the molecular chains are linked together. Both reactions occur in the presence of oxygen, and they are constantly competing with each other—each reaction undoes the work of the other.

The cutting reaction tends to make the rubber soft and tacky, while the cross-linking reaction tends to make it hard and brittle. Whether a rubber softens or hardens at high temperatures depends on which of the two reactions is the faster. In natural rubber, cutting occurs more rapidly than cross-linking at first but later the rate of cross-linking overtakes and passes the rate of cutting. This is the reason that natural rubber softens first and then hardens when it is heated in air at high temperatures. In butyl rubber, cutting is always faster than cross-linking and so butyl softens in heat aging. In GR-S, the main synthetic rubber manufactured by the Government at the present time, the rubber molecules are always cross-linked more rapidly than they are cut, therefore the rubber becomes hard and brittle if heated for any length of time. Buna N type synthetic rubbers also harden in the same way. The good heat-aging properties of neoprene are not due to unreactiveness of the rubber, because both deteriorative reactions occur very rapidly in neoprene. The reason is that the two reactions occur at almost the same rate, and so practically cancel each other.—*From a paper by R. D. Andrews and A. V. Tobolsky, Princeton University, presented at the recent annual meeting of the A.S.M.E. in New York.*

Sealing Against High Pressures

IT WAS FOUND during the development of the General Motors diesel engine and injector that, for the best combination of fuel consumption and clean exhaust for all loads and speeds, the spray tip velocity of the fuel entering the combustion chamber during the injection period should be about 2300 feet per second at 2100 revolutions per minute. Only by the use of such velocities at high speed was sufficient atomization at low-speed full-load operation obtained. Due to the wide speed range of 250 to 2100 revolutions per minute, and also to the fact that the injection pressure varies as the square of the spray tip velocity, resultant injection pressures encountered are of the order of 40,000 pounds per square inch at top speed. These high pressures during the injection cycle impose many difficulties on manufacturing from the standpoint of making the component parts so that they will successfully seal and yet not present a commercial barrier from the standpoint of cost.

The high-pressure seals in this injector are of two types: Flat and cylindrical. The flat high-pressure seals are the bushing lower face, spacer, check valve seat, and spray tip top face. Satisfactory surfaces on these parts are obtained by surface grinding flat and smooth, leaving approximately .001-inch per side for finish, prior to hardening and drawing to rockwell C 62 minimum. After heat treatment the surfaces are rough-lapped to .0001-inch over finished size after which each piece is finished individually by hand lapping optically flat to within one wavelength of sodium light and to a 1½ microinch surface maximum with a pumice-charged lapping plate. Attempts have been made to accomplish this result mechanically, but so far are not wholly successful. When thus prepared and assembled clean, the resultant compressive load applied to the stack of parts in assembly effectively seals the pressures encountered.

Requirements of Cylindrical Seals

The cylindrical high-pressure seal is formed by the plunger and bushing which must fit so closely together that to all intents and purposes none of the high-pressure fuel is allowed to escape from the bushing chamber, yet at the same time, fit so freely that when both parts are thoroughly cleaned, the bushing will spin on the plunger. A seal of this type is considered satisfactory when leakage passing the plunger and bushing does not dilute the crankcase oil over 3 per cent in 500 hours. A minute leakage of this order is so insignificant that it has no practical effect on individual injector calibration.

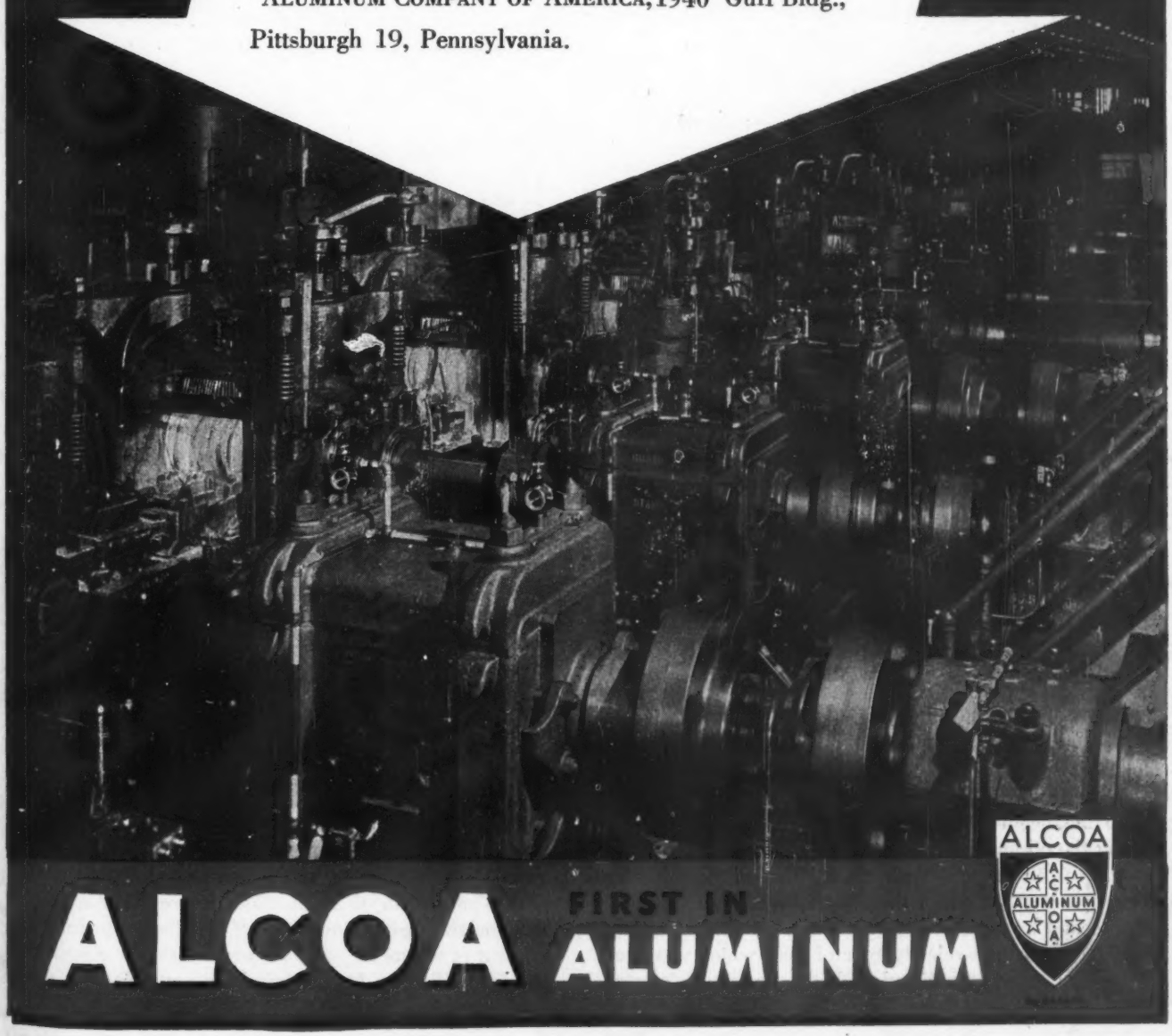
This seal is obtained by internal centerless grinding the bushing carefully to a tolerance of .0005-inch, after it has been hardened and drawn to rockwell C 40 for core properties. The bushing is then nitrided to a hardness of rockwell superficial 82 minimum, using a 30-kilogram load. Approximately .001-inch is then removed by honing to a diametrical tolerance of .0004-inch in any one bushing. This corrects the vast majority of whatever taper, bellmouth, and bow has been left from grinding. The hole is then rough-lapped on special machines using floating bushing holders and cast-iron laps which are adjustable for size by driving them farther on their tapered arbor. On this operation the holes are held to a tolerance of .0002-inch in any one bushing. This operation further corrects for taper, bellmouth, and bow. The final operation on the bushing hole is done by hand-lapping with a lap in a bench reduction motor operating at 550 revolutions per minute, and the holes are then free from

This rolling mill is evidence of Alcoa's faith in the future for Aluminum Wire, Rod and Bar

Started as a prewar project, this rolling mill at Alcoa's Massena, New York, plant was rushed to completion to make it available for war production. Aluminum alloy rod and bar stock rolled here has been an important contributor to the war effort.

As wartime demands slack off, the production of this mill will be available to those manufacturers who plan big things with aluminum postwar.

Alcoa has great faith in the future for aluminum wire, rod and bar. May we tell you why? Write to ALUMINUM COMPANY OF AMERICA, 1940 Gulf Bldg., Pittsburgh 19, Pennsylvania.



ALCOA FIRST IN ALUMINUM

ALCOA ALUMINUM

undesirable taper, bellmouth, and bow. The resultant surface finish is within one microinch. Finished size of the hole is .250 to .251-inch diameter, depending on initial size and amount necessary for correction.

After having been hardened and drawn to rockwell C 45 and nitrided to rockwell superficial 82 minimum using a 30 kilogram load, the plunger is ground round and straight on centers to .0015-inch over a bushing plug, plus or minus .00003-inch for diameter, and held to .00002-inch for taper. It has been found that those close limits more than pay for themselves in subsequent lapping operations. It is then lapped and polish-lapped, enough smaller than the plug to give an operating clearance of 30 to 60-millionths of an inch in the bushing—a working tolerance of 30-millionths of an inch, maximum.

With this technique of manufacture, all close measurements become external and are easily measured on an electrolimit gage. The plunger is a noninterchangeable matched fit with the bushing, and will enter the bushing only when both pieces have been wiped perfectly clean. It is interesting to note that if the surface of the plunger is touched even with clean fingers, it will no longer enter the bushing freely. During the lapping operations, plungers and bushings are lapped separately; never together.—*From a paper by C. W. Truxell Jr., General Motors Corp., presented at the national war materiel meeting of the S.A.E. in Detroit.*

Developing the Jeep Powerplant

ONE reason for the jeep's popularity on landing beaches, over rough fields, and in all climatic conditions is its reserve power. The Willys passenger-car engine, a 4-cylinder, L-head unit with a 3⅛-inch bore and 4⅜-inch stroke, giving a 134.2 cubic-inch displacement, was chosen as the power-plant because it was tooled up for production and had been widely tested in actual service in the hands of owners.

Intensive development work had to be undertaken to make the jeep into a thoroughly reliable and sturdy vehicle. Three examples show what can be accomplished with a design by intensive study of the causes for failure and using every possible resource of material and design that can be employed without changing the basic design.

Example one was the transmission, which at first failed in the sand test at Fort Meade in approximately 3 hours, and which showed bad pitting of teeth on the first-speed transmission gears at the end of 3½ to 4 hours, and a failure of needle bearings in the clutch pinion at the end of 8 hours. These causes of failure were attacked by using floating bushings in the countershaft clutch gear instead of pressed-in bushings, and providing adequate oil to them; by the use of Amola steel in the cluster gears instead of nickel steel; by making the constant-mesh clutch pinion gear and cluster gear act as an oil pump to supply oil to the needle bearings in the clutch pinion; and by raising the oil level in the transmission. Transmission life was increased 10 to 1 with these minor changes, lifting the whole gearbox out of the danger zone.

A second example was given in the intensive study of the Tracta constant-velocity type of joint which was used in the front axle as optional with the Bendix-Weiss constant-velocity joint. Here it was found that stress raisers existed in certain sections of the forked ends of the drive shafts, which caused failure in from 27,000 to 70,000 cycles on the fatigue test set-up. By actually reducing some of the section where the stress raisers existed, spreading out the stressed area so that the stresses were less concentrated, and by shot blasting, the life of these members was raised to 200,000 cycles without failure, which was considered adequate.

A third example was the development of the jeep engine itself which, over a period of two years, underwent intensive study in every detail, resulting in changing the engine from one which gave 45 horsepower at 3200 revolutions per minute and an extremely short life on block test at wide-open throttle and high speed, to an engine which regularly undergoes a life test of 4400 revolutions per minute at wide-open throttle for 100 hours without failure. These tests and studies covered changes from iron to aluminum pistons; a study of cylinder-head sections and stresses, pressure as well as water distribution; a study of main bearing loads and the effect of flywheel mass and counterweighting; an intensive study of cam design and valve-spring design; a study and alterations of the method of feeding oil into the bearings and rods; and finally a careful analysis of the pneumatics of the engine. All of these changes preserved the original tooling of the engine and were made without interrupting its production.—*From a paper by D. G. Roos, Willys-Overland Motors, Inc., at a recent meeting of the Metropolitan section of the S.A.E.*

Oxyacetylene Pressure Welding

IN THE new oxyacetylene pressure-welding process, which may be fully or partly mechanized, the parts are butted under nominal pressure, heated by means of multiple small oxyacetylene flames to a temperature of about 1200 degrees Cent. and upset to a controlled degree. The process has special advantages over currently used mechanized welding methods, among which are: Adaptability to those high-carbon and alloy steels that are welded with difficulty by fusion-welding methods; smoothness and uniformity of the completed weld; short welding time, and relatively low unit cost. Physical properties of these pressure welds are excellent.

Technically, oxyacetylene pressure welding is known as "solid phase welding" because it is a method of joining two structural members in which none of the material is melted at any time during the welding process. As a result of experiment it has been found that the bonding mechanism or method by which a sound, strong weld is obtained, without the melting and fusion usually associated with the formation of a weld, comprises atom transfer across the interface (contacting surfaces of abutting members) in accordance with the laws of diffusion, with a resultant trans-interface crystallization. Temperature is found to be the dominant factor in the practical application of solid phase welding. A temperature of 1050 degrees Cent. appears to be the feasible minimum for steel, with a maximum of about 1200 degrees Cent., which is well below the melting point of the lowest melting constituent of the metal—*From a paper by A. B. Kinzel, vice-president, Electro Metallurgical Co., presented at the recent annual meeting of the A.S.M.E. in New York.*

Will you join us in the conning tower?

WHAT is the future of your product after peace comes? How keen will your competition be, and what can you do to prepare against it? Because we have been privileged to assist in the improvement of many war machines by means of forgings, we have gained a fund of information you will find worth sharing.

Forged aluminum cylinder heads, forged aluminum pistons, and steel cylinder barrel forgings for aircraft engines are among the Tube Turns achievements which have been called revolutionary. Produced to highest precision standards, in large quantities, with accurate heat treating under rigid laboratory control, these products point the way to a wide range of possibilities for peacetime industry.

Whether you are concerned with the improvement of your present product, or with developing wholly new ideas, the successful experience of Tube Turns engineers is available. TUBE TURNS (Inc.), Louisville 1, Kentucky.

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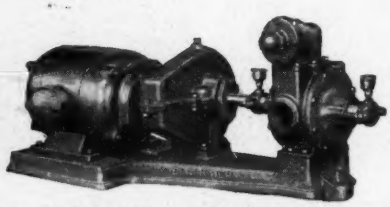

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Capacities 5 to 750 GPM. Pressures to 500 psi. Temperatures up to 600°F. For pumping liquids varying in viscosity from butane to asphalt.

Direct-connected pump.

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Capacities from 1-1/2 to 25 GPM. Pressures to 125 psi. Vertical or horizontal mounting.

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Swinging vane principle makes these pumps SELF-ADJUSTING FOR WEAR. When the buckets finally wear out, a 20-minute replacement job restores the pump to normal capacity.

NATION-WIDE ENGINEERING SERVICE

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No. 306: Facts about Rotary Pumps.
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BLACKMER PUMP COMPANY
1970 CENTURY AVE. GRAND RAPIDS 9, MICH.

Applying Fatigue Data

(*Concluded from Page 146*)

$$A=\left(1+\frac{ec}{r^2}\right)\left(\frac{P_{max}}{S_u}\right)\times$$

$$\left[\frac{1}{kp}-\left(\frac{P_m}{P_{max}}\right)\left(\frac{1}{kp}-1\right)\right] \quad \text{.....(12)}$$

or

$$A=Q\left(1+\frac{ec}{r^2}\right)\left(\frac{P_{max}}{S_u}\right) \quad \text{.....(13)}$$

where

$$Q=\frac{1}{kp}-\frac{P_m}{P_{max}}\left(\frac{1}{kp}-1\right)$$

The values of Q, as explained in Example 1, represent the ordinates to the diagrams in *Fig.* 7. *Fig.* 7 can be used, therefore, for the design of eccentrically loaded members, since with Q known Equation 13 can be used for finding the cross-sectional dimensions of the member.

In the application of the foregoing equations the designer must not only obtain fatigue stress data but he must decide what values to use for the factor of safety. It is known that the values for the factor of safety depend upon many factors, such as the possible errors in the stress analysis for the particular machine member, the extent of the variations in the material composition, the influence of the fabrication methods used, certain time effects such as corrosion, and the seriousness of possible failure of the member. There are also certain factors influencing the fatigue strength which the designer must consider in selecting the factor of safety.

Factors Influencing the Fatigue Strength

There are many test variables that influence the fatigue strength of materials. A book by the Battelle Memorial Institute staff(5) gives a comprehensive discussion of these. Some of the influences are:

1. **Influence of the number of stress cycles used for defining fatigue strength:** A value of $N = 10^7$ has been used commonly for ferrous metals and alloys, whereas a higher value of N is used for nonferrous alloys since often the stress cycle relation does not approach an asymptote. It should be noted, however, that the value of N selected depends upon the estimated life of the member to which the test data are applied.

2. **The effect of rapidity of stress applications:** If the speeds used are very high, their influence on the fatigue strength must be considered, otherwise the influence of speed of testing is negligible.

3. **Influence of cold-working:** There have been several instances where cold working has been used to increase the fatigue strength of a member.

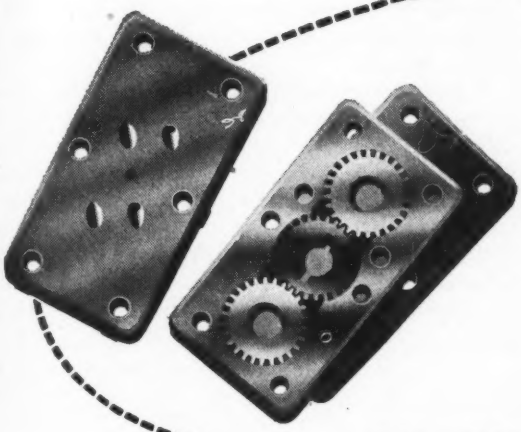
4. **Effect of heat treatment:** Special heat treatments have been used to increase fatigue strength.

5. **Influence of residual stresses:** Although the magnitudes of this influence are uncertain, the effect of residual stresses is an important consideration in the design of some machine parts.

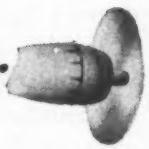
6. **Effect of overstressing and understressing:** By un-

ANOTHER MASS-PRECISION ASSEMBLY BUILT BY NICHOLS

Putting the "No Ice" Sign



on an Airplane's Propeller



A propeller blade is certainly no place for an ice-pack. Neither is a pilot's windshield. Yet every drop of freezing rain that strikes these surfaces is potentially ice-forming... and dangerous.

W. H. Nichols & Sons has helped combat this natural "enemy" by producing a unique precision pump for an anti-icing unit which "beats ice to the draw" by spreading a thin film of alcohol on these exposed surfaces. The mixture of alcohol and rain with its lower freezing point quickly flies off as harmless vapor instead of turning to ice.

Metering such a thin fluid as alcohol with *exact, measured* distribution to both engines in a bi-motored plane presents many unusual manufacturing requirements. The pump must operate *completely without lubricant* to insulate the rubbing surfaces from one another. This means fine, flat surface finishes and strictest dimensional accuracy. Since the whole unit is powered by the plane's limited electrical supply, complete freedom from binding is essential.

Nichols mass-produced the close dimensions required...gears thicknessed to plus 0" minus .0001"...side plates flat to .0002" over 3½" length...bores in the centerplate held to plus .0002" minus 0". A high production rate was constantly maintained *without pre-selection of any parts* and the cost kept low.

Such performance is typical of Nichols' forty years of mass-precision manufacturing ability. Here unique facilities have been developed for producing the "hard" parts.

Perhaps you have an unusual precision problem—part or assembly—which can benefit from such a tradition of accuracy in mass-production. Talk it over with "Accurate" Nichols.

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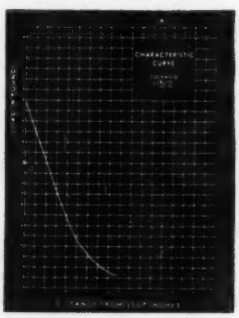
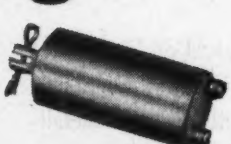
With Cannon Electric D. C. Solenoids you get automatic trigger-action. Solenoid pull is positive. Cannon Solenoids may be operated without damage to a maximum temperature of 200 degrees C.

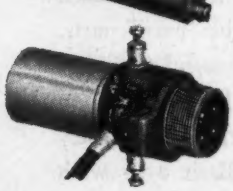


Response characteristic charts like the above (test at 28 volts) are found in Solenoid Bulletin.

A wide selection of Cannon Solenoids are built for various combinations of voltages, pull, armature travel, duty cycle, etc. to suit design requirements. Nominal voltages are 12 and 24; others by special order. Eight types have "holding" features.

As a reliable means of controlling or operating various devices, they are used for retarding magnetos in starting engines, operating hydraulic valves and mechanical clutches, firing machine guns, arming and releasing bombs, dropping auxiliary fuel tanks, locking retracting gears, film-winding mechanisms and for shutter control on special cameras. These are only a few of the possible applications.





A copy of the Cannon Electric Solenoid Bulletin—34 pages of diagrams, data and photos of the complete line will be sent free for the asking. Address Dept. A-107, Cannon Electric Development Co., 3209 Humboldt Street, Los Angeles 31, California.

Also manufacturers of Electrical Connectors

CANNON ELECTRIC

Cannon Electric Development Co.
Los Angeles 31, Calif.

Canadian Factory and Engineering Office:
Cannon Electric Company, Limited, Toronto

Representatives in Principal Cities—
Consult Your Local Telephone Book

derstressing a specimen is meant subjecting it to repetitions using a safe stress range before testing at a higher stress range. Many experiments show that the fatigue strength is increased by such a treatment. On the other hand, initial overstressing of a specimen decreases the fatigue strength.

7. Influence of high and low temperatures: With high temperatures there is a decrease in fatigue strength, while in some cases at low temperatures there is an increase in fatigue strength.

8. Effect of surface condition: The presence of corrosive action may appreciably affect the fatigue strength. The smoothness of surface and the methods used in machining are also important in their effect on fatigue strength.

9. Influence of stress concentration: The machine designer must constantly avoid the use of any sharp discontinuities in sections and dimensions of parts and use fillets with ample radii to avoid high stress concentration. Fatigue stress concentration factors have been obtained(6) for some of the common type discontinuities such as bars with holes, and fillets and keyways subject to reversed axial and bending loads.

In selecting the factor of safety the machine designer must consider the foregoing influences and, if possible, estimate the magnitude of these influences on the fatigue strength.

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   H. F. Moore and G. N. Krouse—University of Illinois, Engineering Experiment Station, Circular 23, 1934.
4. J. O. Smith—*The Effect of Range of Stress on the Fatigue Strength of Metals*, Bulletin No. 334, Engineering Experiment Station, University of Illinois, 1942.
5. Battelle Memorial Institute Staff—*Prevention of the Failure of Metals Under Repeated Stress*, John Wiley & Sons, Inc., New York, 1941.
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Zinc Restrictions Removed

Restrictions have been modified on the use of zinc and zinc products as follows: Washing machines, vacuum cleaners, sweepers and such equipment has been removed from the prohibited list and the use of zinc therein is permitted when production is restored. Other items on List A such as automotive trucks, tractors, diesel engines, gas-fired stoves and ranges now being manufactured in limited quantities may use zinc if the manufacturers desire and if there is no restriction in any other WPB order. Restrictions on the use of zinc or zinc products for protective coating or plating (other than paint) except for articles on List A, are removed. List B has been made up enumerating articles for which zinc or zinc products may be used, and for articles not on either Lists A or B, the quota has been increased from the previous 60 per cent to 80 per cent for zinc products. Relief from provisions of this order M-11-b, are limited to such action as may be authorized under Priorities Regulation No. 25 which provides for "spot" authorization where it does not interfere with war production.

Valuable idea here for product designers

Magnesium Core Boxes are Easier on Womanpower

The lighter weight of magnesium core boxes makes them easier to lift and manipulate. Women core makers, therefore, find their work less tiring. Much of the usual fatigue slump is avoided. Foundrymen report that magnesium core boxes stand up well in this service.

a liftable idea!

Properties making magnesium so attractive to foundry operators should make it equally valuable for you. Lighter weight magnesium tools and equipment require less effort to do a job. Tasks are more to labor's liking.

Operators of machines with moving parts of magnesium find its lighter weight similarly desirable. Because of lower inertia, those machines can be safely run at higher speeds. There's less tendency to vibrate, less wear and tear. Upkeep is reduced. Production is increased. Quality of products is higher.

May we help you adapt magnesium to your uses? Aluminum Company of America, 1703 Gulf Building, Pittsburgh 19, Pa.

MAGNESIUM MAZLO PRODUCTS

AMERICAN MAGNESIUM CORPORATION

SUBSIDIARY OF ALUMINUM COMPANY OF AMERICA

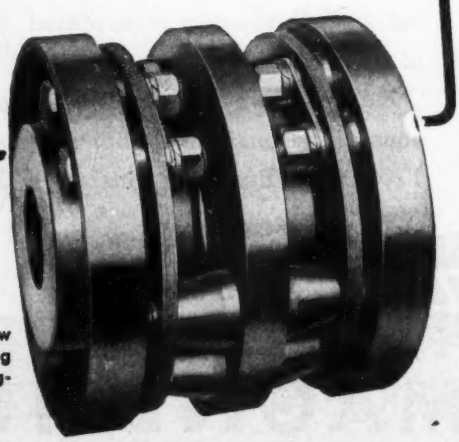
THOMAS
FLEXIBLE
COUPLINGS

THE most dependable couplings ever made, Thomas Flexible Couplings with their five exclusive features save millions of dollars in maintenance every year and provide longer life for your equipment.

★

NO BACKLASH
NO WEAR
NO LUBRICATION
NO THRUST
FREE END FLOAT

These are the five essential features for a permanent care-free installation not found in any other make or type of flexible coupling.



Write for the new Thomas Catalog with complete engineering data.

"BUILT TO LAST A LIFETIME"

THOMAS FLEXIBLE COUPLING CO.
WARREN, PENNSYLVANIA

F-2

BUSINESS AND SALES BRIEFS

ELECTION of George F. Farley as vice president in charge of sales for Spincraft has been announced by Milwaukee Metal Spinning Co., Milwaukee. Mr. Farley will be responsible for all sales activity, product development and post-war planning.

♦

Formerly manager of the Seattle office, A. J. Schmitz has been named Pacific regional manager for Allis Chalmers Mfg. Co. with headquarters in San Francisco. He has been associated with the company since 1914.

♦

Allegheny Ludlum Steel Corp. has purchased property at 209 Beaumont street, St. Louis, from National Refining Co. A district office and warehouse have been established at this address.

♦

Because of the present widespread use of fractional horsepower motors, Bodine Electric Co. has started construction of a two-story addition to the present plant facilities at 2254 West Ohio street, Chicago. This will increase the present capacity approximately 50 per cent.

♦

Celebration of its tenth anniversary has been announced recently by Superior Tube Co., manufacturer of seamless and Weldrawn small metal tubing.

♦

Victor Failmezger, formerly application and field engineer, has been made sales manager of Quimby Pump Co. of Newark and New Brunswick, N. J., a subsidiary of H. K. Porter Co. Inc., Pittsburgh.

♦

Recently announced is the appointment of John G. Davison of Denver as Colorado representative for Wheelco Instruments Co., Chicago. Mr. Olin Phillips of Toledo has been made representative in the Northern Ohio territory. The state of Minnesota has been added to the territory covered by L. E. Cyrtmus of Milwaukee.

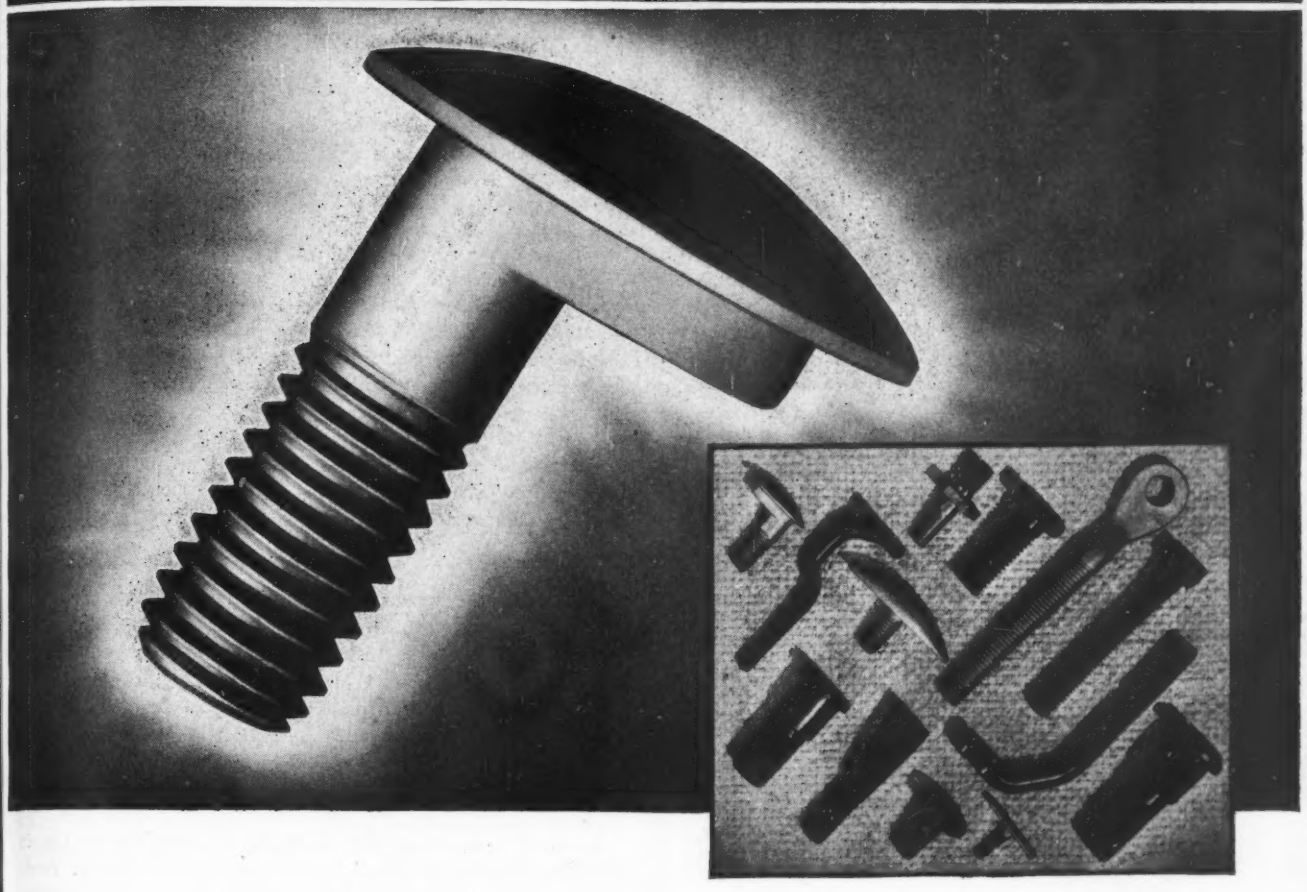
♦

Plans to double the capacity of the Foamglas plant at Port Allegany, Pa., have been made by Pittsburgh Corning Corp. in order to handle increasing demands for cellular glass insulation.

♦

Three appointments in the Motor division have been announced by General Electric Co. Elliott Harrington, manager of sales of Schenectady induction motors, has been made manager of sales of the newly formed integral-horsepower, alternating-current motor section in Schenectady. In charge of the direct-current motor group since 1936, J. T. Farrell

100 YEARS *Making strong the things that make America strong*



How would you have it made . . .

for maximum strength? . . . for utmost accuracy? . . .
for best appearance? . . . for lowest cost? . . .

In every case, the answer would be **"By COLD-FORGING"**

Product strength is maximum because cold-working increases the tensile strength of the metal and also serves as an automatic inspection of the material.

Accuracy and appearance of the cold-forged product are superior. Modern cold-upsetting equipment provides the maximum in precision and fine finish. Billions of aircraft fastening parts have been made by this process.

Costs are held to a minimum, both by the high speed of the cold-forging equipment and by the saving in raw material through practically complete elimination of scrap.

When your planning calls for a product or part shaped like any of those at the right, consider cold-forging, which may be not only the *best*, but often the one efficient method of production. As soon as present wartime production requirements are fulfilled — RB&W will be able to give you the benefit of its high-production facilities and long experience — *longest experience in automatic cold-forging.*

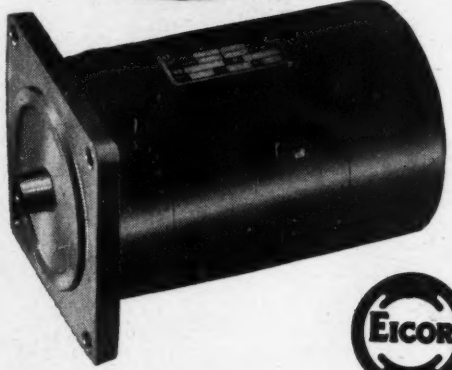
RB&W

Russell, Burdsall & Ward Bolt and Nut Company . . . with factories at Port Chester, New York, Coraopolis, Pa., Rock Falls, Ill., . . . with sales offices at Philadelphia, Detroit, Chicago, Chattanooga, Los Angeles, Portland, Seattle . . . with the industry's most complete, easiest-to-use catalog.

EMPIRE BOLTS RIVETS AND ALLIED FASTENING PRODUCTS · SINCE 1845

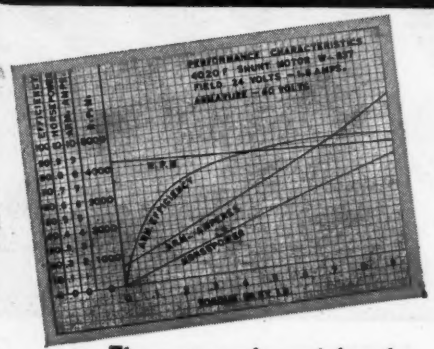
RUSSELL, BURDSALL & WARD BOLT AND NUT COMPANY

MOTOR DATA
No. 129



EICOR

4000 FRAME MOTOR
1/2 HP at 3900 RPM



The output—the weight—the size—of these 4000 Frame Motors are features well worth remembering. Every adaptation of the standard design is engineered for the precise requirements of an aircraft, portable, or industrial application.

FEATURES

ELECTRICAL
Series, shunt, or compound-wound
Unidirectional or reversible
Optional torque
Optional speed
Optimum efficiency
For control circuits
Electric braking optional

MECHANICAL
Ventilated or enclosed types
Base or flange mounting
Operation in any position
Low space factor
Ball bearing equipped
Optional shaft details
Rugged construction

4000 FRAME MOTORS		4020 Shunt	4020 Series
Watts, Output, Con.	(Max.)	375	746
Torque at 3900 RPM	(ft. lbs.)	.65	1.4
Torque at 6000 RPM	(ft. lbs.)		.88
Speed Regulation		8%	
Lock Torque	(ft. lbs.)	2.5	4
Volts Input	(min.)	12	24
Volts Input	(max.)	110	110
Diameter		4"	4"
Length Less Shaft		7⅛"	7⅛"
Shaft Dia.	(max.)	.625"	.625"
Weight	(lbs.)	9.2	9.2

EICOR INC. 1501 W. Congress St., Chicago, U. S. A.
DYNAMOTORS • D. C. MOTORS • POWER PLANTS • CONVERTERS
Export: Ad Auriemo, 89 Broad St., New York, U. S. A. Cable: Auriemo, New York

has been appointed manager of sales of the newly formed integral-horsepower direct-current motor section which also is located in Schenectady. D. A. Yates has been named assistant manager of sales of both sections in charge of Lynn motor sales group and will make his headquarters at the Lynn, Mass., River Works.

♦

Kennametal Inc., Latrobe, Pa., has removed the Detroit branch to larger offices at 5531 Woodward avenue. Bennett Burgoon Jr., who represented the company at Rockford, Ill., for several years, has been named district manager of the new office.

♦

Reopening of the Chicago sales office in the Engineering building, 205 West Wacker drive, has been announced by The Beryllium Corp. of Pennsylvania. Thomas E. Neal, graduate engineer, has been named district manager.

♦

Creative Plastics Corp., manufacturer of terminal strips, bushings, etc., has appointed Allied Radio Corp., Chicago, as exclusive midwestern distributor.

♦

According to a recent announcement by Wickwire Spencer Steel Co., the general sales offices of the Springs and Formed Wire division and the Automotive division have been moved to New Bond street, Worcester, Mass. Also located at Worcester is the Morgan plant at which the products of these divisions are manufactured. A district sales office of the divisions will be continued at 500 Fifth avenue, New York 18.

♦

Previously sales manager of Bright Light Reflector Co. in Brooklyn, Charles E. Scholl has joined Federal Electric Products Co. Inc. as general sales manager. In his new position Mr. Scholl among other duties will direct all sales and take charge of warehousing in principal cities from coast to coast.

♦

Succeeding S. H. Gorham is W. A. Meyer as manager of dealer sales for Allis-Chalmers Mfg. Co. Mr. Meyer has been associated with the company since 1926.

♦

Opening of a sales office at 108 North Main street, South Bend, Ind., has been announced by Cutler-Hammer Inc., Milwaukee. Terry Fisher, South Bend representative, has been placed in charge of the new office.

♦

Connected with the company since 1932, Kenneth McCreary has been elected president of Goetze Gasket & Packing Co. Inc., New Brunswick, N. J. In this position Mr. McCreary succeeds Frederick Goetze.

♦

According to a recent announcement, Fred W. Dixon Jr. has joined Cleveland Foundry Co. as sales manager. Mr. Dixon had served as sales engineer and assistant sales manager of the Gunite Foundry in Rockford, Ill., for more than twelve years.

♦

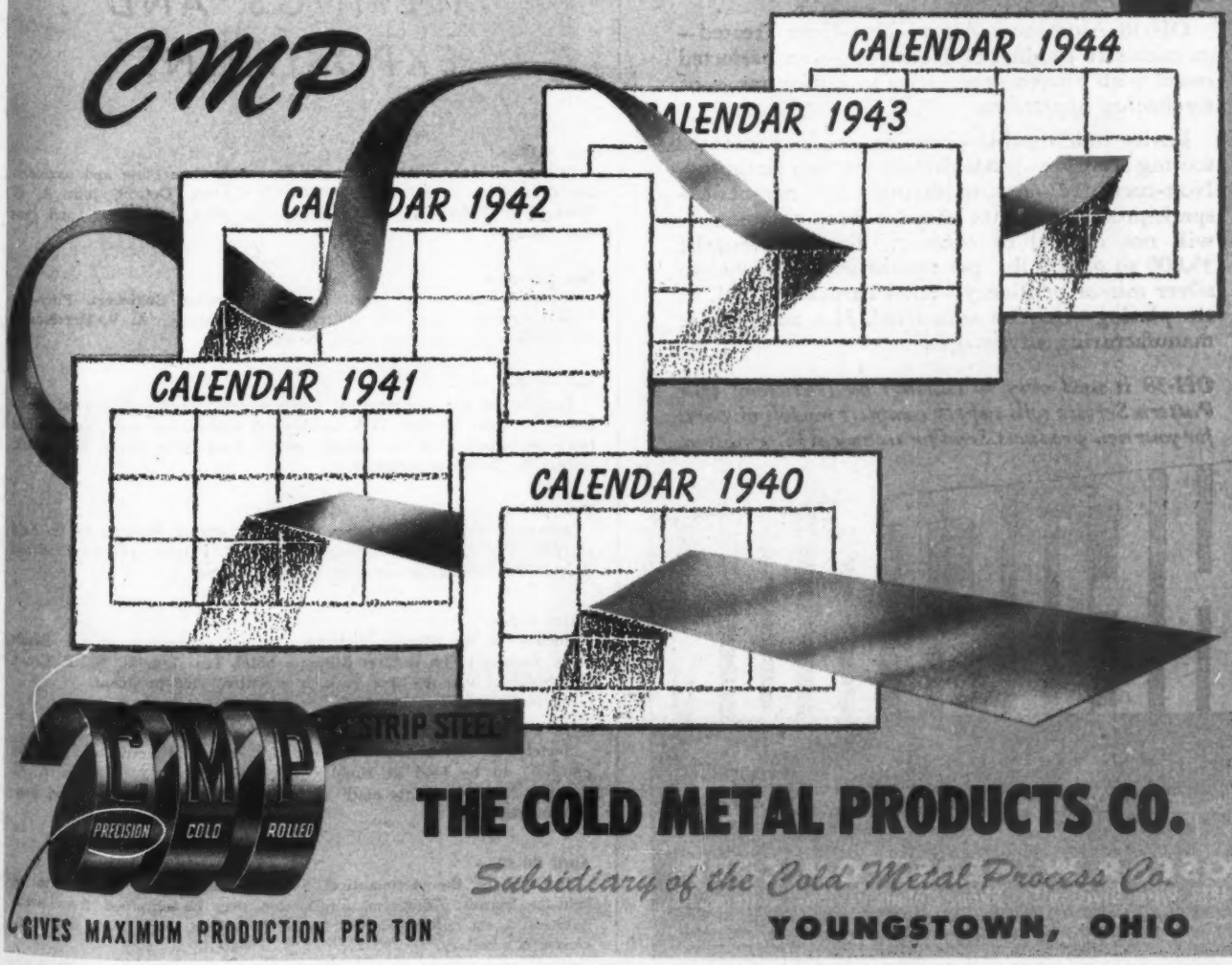
Appointment of the following three field engineers for the sale of heat and corrosion-resistant alloy castings has been announced by Sterling Alloys Inc.: John Sonnenfeld, 206 North Seventh street, Keokuk, Iowa, will cover Missouri, Western Iowa, Southern Illinois, Kansas and Nebraska; A. C. Woolley,

Tolerance Today .0002
in Cold Rolled Strip Steel

● Every year brings new achievements in CMP precision cold rolling of strip steel. Most recent and most remarkable of our precision operations is the rolling of miles of war-important steel strip with tolerance of .0002 in. Spurred by the urgency of the need, CMP undertook what authorities described as humanly impossible: to maintain a practically zero tolerance in cold rolled high tensile metal. Neither men nor machines were known able to "mike" unfailingly such extremely close tolerances. CMP is doing it with specially constructed equipment, and the armed forces now have one more precision aid to Victory.

CMP gets the call whenever absolute control of physicals and dimensions must be guaranteed. First to specialize in precision cold rolling, CMP now has experience and equipment with a potential of even greater accomplishment. CMP precision cold rolled strip steel, with its wide range of physicals, choice of finishes, tempers, grades, widths and gauges, gives greater yield of parts per ton and protects intricate dies and machinery because each shipment is tailored to a specific job. Write for a generous test sample.

PIONEERS OF PRECISION IN COLD-ROLLED STRIP STEEL



THE COLD METAL PRODUCTS CO.

Subsidiary of the Cold Metal Process Co.

GIVES MAXIMUM PRODUCTION PER TON

YOUNGSTOWN, OHIO

"OH-38"

WILL HELP YOU DO IT IN 1945

Sustained post-war employment means large volume production of better products at reasonable prices . . . that's the story.

● Many component parts of your post-war products can be made better, faster, cheaper—with OH-38.

OH-38 Aluminum Alloy—Non-Heat Treated—an exclusive product of Hedstrom—is a perfected metal with proven superiority in a wide range of machining operations.

Easily machinable—easy on tools—increases tooling accuracy—holds threads without stripping. Non-corrosive—non-oxidizing—non-magnetic—spark-proof—does not require heat treatment—will not expand or contract. Tensile strength: 35,000 to 40,000 lbs. per square inch. Polishes to silver mirror brilliancy—takes chrome, nickel, or tin plating—may be annodized. Has many other manufacturing advantages.

OH-38 is used only in castings by Hedstrom. Our Pattern Service will supply complete models of parts for your new products. Send for technical information.

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4818 West Division St., Phone Columbus 3667, Chicago, Ill.
Manufacturers of Aluminum, Brass, Bronze and High Conductivity Copper Castings. Pattern Makers. Complete Mechanical Assemblies and Models to Specifications

Terminal Sales building, Portland, Oreg., will handle Oregon and Washington; and G. Dell, 314 West Bloom street, Louisville, Ky., will cover Kentucky, and Southern Indiana including Evansville.

•

All commutators and other manufactured items previously handled by Han-Kor Inc. at 5005 Euclid avenue, Cleveland, will now be handled by Han-Kor Sales Co. Inc. located at 1847 West Bethune avenue, Detroit.

•

James E. Norcross has been appointed sales manager by Arcos Corp., Philadelphia, which produces special welding electrodes.

•

According to a recent announcement, Hercules Electric & Mfg. Corp. is now located at its new plant and offices at 2500 Atlantic avenue, Brooklyn 7, N. Y., where expanded facilities will better fulfill requirements.

•

Plans for the erection of a modern plant on the Pacific Coast for the manufacture of laminated plastics—sheets, rods and tubes—as well as various coated special materials have been announced by Taylor Fibre Co., Norristown, Pa. While equipment has been ordered, the exact location of the new plant has not yet been determined.

MEETINGS AND EXPOSITIONS

Jan. 8-12—
Society of Automotive Engineers Inc. Annual meeting and engineering display to be held at Book-Cadillac hotel, Detroit. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Jan. 22-24—
American Society of Heating and Ventilating Engineers. Fifty-first annual meeting to be held at Hotel Statler, Boston. A. V. Hutchinson, 51 Madison avenue, New York, is secretary.

Jan. 30-Feb. 1—
Institute of the Aeronautical Sciences Inc. Thirteenth annual meeting to be held in New York. Additional information may be obtained from headquarters of the institute at 30 Rockefeller Plaza, New York. Robert R. Dexter is secretary.

Feb. 26-March 2—
American Society for Testing Materials. Spring meeting to be held at Hotel William Penn, Pittsburgh. Robert J. Painter, 260 South Broad street, Philadelphia, is assistant to the secretary.

March 6-7—
Society of the Plastics Industry. Annual conference of the Pacific Coast Section to be held at Biltmore hotel, Los Angeles. W. T. Cruse, 295 Madison avenue, New York, is executive vice president.

April 4-6—
Society of Automotive Engineers Inc. National aeronautic meeting (Spring) to be held at Hotel New Yorker, New York. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

April 26-27—
Institute of the Aeronautical Sciences Inc. Spring meeting to be held in Detroit. Additional information may be obtained from headquarters of the institute at 30 Rockefeller Plaza, New York. Robert R. Dexter is secretary.

When there's "no other way"... look to

AMERICAN SEAMLESS FLEXIBLE TUBING!

THE AIR LINES of this Vaill No. 6 Tube Beading and Flaring Machine are made of American Seamless Flexible Tubing. Four lengths of ⅜" tubing connect two ports on each of two cylinders with the air control valve. A fifth length of ½" tubing carries the main air supply to the valve.

Mr. John L. Vaill, president of the Vaill Engineering Company, points out that, while there is no movement in the flexible tubing once it is installed, lack of space made it impossible to make the connections with rigid pipe. The use of preformed copper tube was also considered but discarded as impractical. American Seamless Flexible Tubing, as illustrated, is installed regularly in these units in less than an hour's time.

Whenever you have a problem in conveying liquids or gases, in connecting moving or misaligned parts, isolating vibration or in shielding electric wiring, remember that there is an American Metal Hose product that will solve your problem. Assistance of our Technical Department is available on request. Write for our new 40-page Catalogue, SS-50, describing Seamless Flexible Tubing.

44198

AMERICAN METAL HOSE BRANCH
OF THE AMERICAN BRASS COMPANY
Subsidiary of Anaconda Copper Mining Co.
General Offices: Waterbury 88, Conn.
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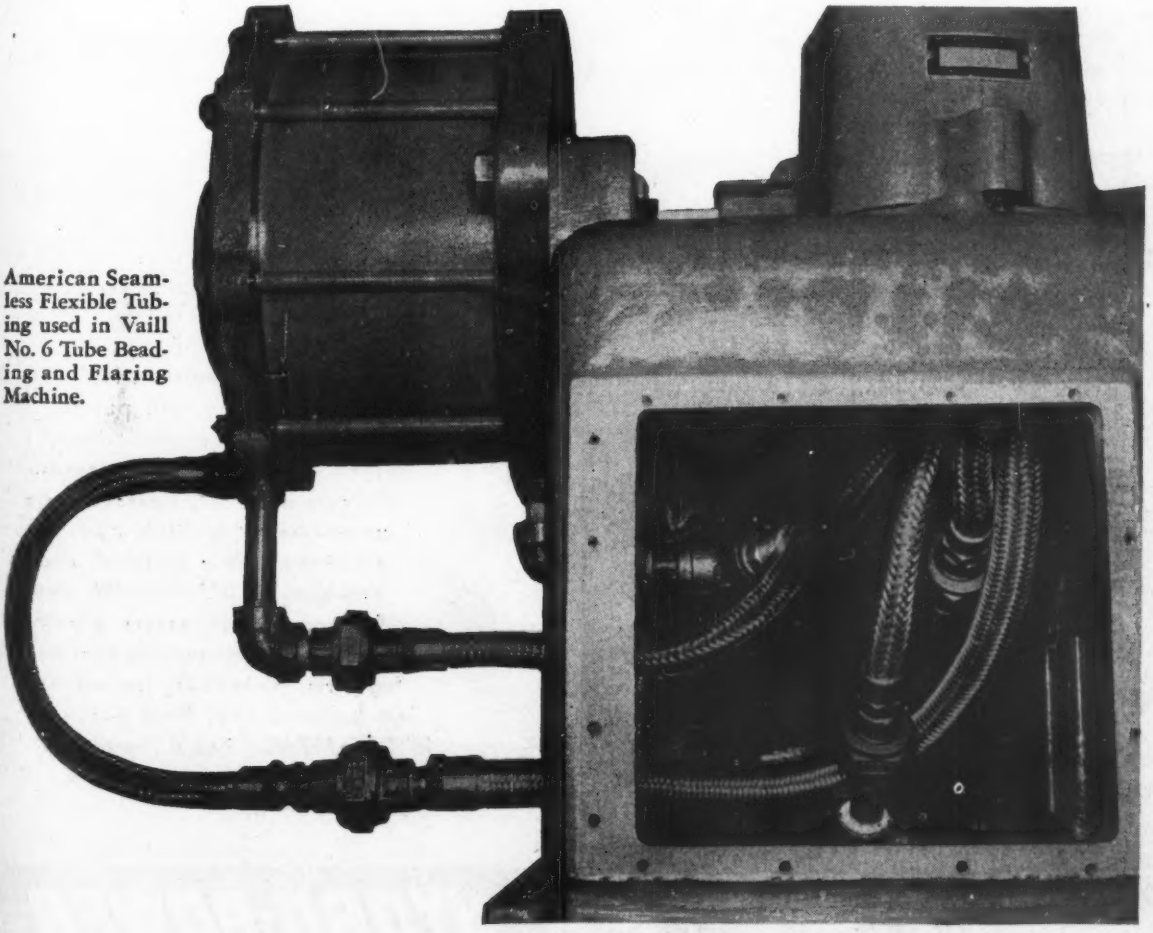

American Seamless Flexible Tubing used in Vaill No. 6 Tube Beading and Flaring Machine.

ANACONDA

American Metal Hose

NEW MACHINES—

And the Companies Behind Them

Armament

Machine for reaming nose end of shells, Davis & Thompson Co., Milwaukee 14.

Cleaning

Precision ball bearing cleaning machine, L & R Mfg. Co., Arlington, N. J.

Food

Foodstuff and materials disintegrator, Franklin McAllister Corp., Joliet, Ill.

Mobile canteen, S. Blickman Inc., Weehawken, N. J.

Industrial

*Internal gear checker, National Broach & Machine Co., Detroit.

Photometer for measuring light transmission, General Electric Co., Schenectady, N. Y.

Electronic magnetic fluxmeter, The Hickok Electrical Instrument Co., Cleveland.

Portable centrifugal pump, Brady Pump Co., Muncie, Ind.

Materials Handling

Combination crane truck, The Elwell-Parker Electric Co., Cleveland.

Lubrication

Portable coolant and cutting oil handling and cleaning machine, W. R. Carnes Co., Madison, Wis.

Metalworking

*Multipurpose, oil-hydraulic bench press, The Denison Engineering Co., Columbus, O.

Horizontal 3-way machine for boring, Le Maire Tool & Mfg. Co., Dearborn, Mich.

Hydraulic press for automobile rims, E. W. Bliss Co., Brooklyn 32, N. Y.

Lightweight air grinder, Madison-Kipp Corp., Madison, Wis.

Cable swager, Standard Machinery Co., Providence 7, R. I.

Automatic benders, Pines Engineering Co., Aurora, Ill.

Automatic machine for squaring, burring, flaring and beading ferrous and nonferrous tubing, Leonard Precision Products Co., Garden Grove, Calif.

Mining

Magnetic separator, Stearns Magnetic Mfg. Co., Milwaukee.

Powder Metallurgy

Hydraulic press for briquetting parts, The Watson Stillman Co., Roselle, N. J.

Packaging

Automatic assembler, The Package Machinery Co., New York.

Textile

Combination brushing and folding machine, Rich Bros. Inc., Somerville, Mass.

Small capacity spray dryer, Western Precipitation Corp., Los Angeles.

Batch-type shredding and mixing machine, Northmaster Div., Struthers-Wells Corp., Titusville, Pa.

Tricot dryer, Proctor & Schwartz Inc., Philadelphia 20.

Testing

*Fatigue testing machine, Sonntag Scientific Corp., Greenwich, Conn.

Mobile rectifier unit for magnetic testing, W. Green Electric Co., New York.

Welding

*Rocker arm spot welder, Sciaky Bros., Chicago 38.

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*Illustrated on Pages 154-157.

PRESSURE GENERATORS FOR OIL HYDRAULIC CIRCUITS
HYDRO-POWER Radial Pumps have established the reputation for dependable service. All pump members are of sturdy proportions, finished with very close tolerances. Minimum operating clearances are maintained for high hydraulic efficiency. This is practical, due to exclusive HYDRO-POWER design features, which assure precision relationship between the vital working parts. Write today for engineering bulletin 411. Find out why HYDRO-POWER Radial Pumps will best solve your hydraulic pressure generating problems.
There is a HYDRO-POWER Radial Pump for every hydraulic requirement, regardless of size. HYDRO-POWER Radial Pumps are built with output capacities from 1¼ to 185 gallons per minute; operating pressures to 2500 lbs. per sq. inch.
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HYDRAULIC PUMPS AND CONTROLS - VALVES - CYLINDER AND RAM ASSEMBLIES - POWER UNITS - SYSTEMS - SPECIAL HYDRAULIC EQUIPMENT
HYDRO-POWER SYSTEMS, INC.
Mount Gilead, Ohio, U. S. A.